NOTES ON THE

CONSTRUCTION AND WORKING

OF PUMPS.

BY

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PREFACE TO FIRST EDITION.

THE Author has written the series of articles, now presented in book form, for users rather than for makers and students of pumping machinery. He has therefore considered the leading types of pumps and pumping engines as users receive them, or should receive them—*i.e.*, in the finished state. But such particulars are given concerning construction and capabilities, and as far as possible of results actually obtained, as will, it is hoped, be of assistance to engineers, manufacturers, and others, in the difficult task of making the best selection for their service from the many proposals they may have before them.

The earlier chapters deal with matters common to all pumps, such as the conditions affecting the limit of suction lift, piston or plunger speeds, air vessels, pipes and other connections, and with types of valves, packings, and other details.

The Author desires to thank the various makers who have kindly supplied blocks, particulars as to tests, or other information concerning their machinery.

E. C. R. M.

13, Temple Street,
Birmingham,
May 1st, 1902.

PREFACE TO SECOND EDITION.

THIS edition, whilst containing all that appeared in the first edition—with but the substitution in some cases of illustrations of later types of machines—treats also, in the additional Chapters XVI. to XIX., of typical examples of present practice in pumping machinery, and of the results obtained therewith.

Recent years have seen great developments in centrifugal pumps, as a result of which such machines are now available for high-lift services hitherto thought quite beyond them. Considerable space has accordingly been devoted to the treatment of high-lift centrifugal or turbine pumps, and in the appendix there will be found a list of recent Patents that have been granted thereon.

The Author desires to take this further opportunity of heartily thanking the makers who have been good enough to place blocks and particulars at his disposal.

E. C. R. M.

13, Temple Street,
Birmingham,
June, 1907.

CONTENTS.

CHAPTER XIII.—Waterworks, Mine, and other Pumping Engines of Large Capacity.

CHAPTER XIV.—Hydraulic Rams and Windmill Pumps.

CHAPTER XV.—Manual and Animal Power Pumps.

PART II.

CHAPTER XVI.—High-duty Pumping Engines—Present Practice.

CHAPTER XVII.—Types of Reciprocating Pumps for Various Services.

CHAPTER XVIII.—Centrifugal Pumps: Turbine Pumps.

CHAPTER XIX.—Air-lift and other Pumps.

NOTES ON THE

CONSTRUCTION AND WORKING OF PUMPS.

CHAPTER I.

Introduction.

A pump may be defined as a machine by and through which a fluid can be made to flow against the action of gravitation or other opposing force.

With most pumps such flow is brought about by the displacing action of a piston or plunger moving within a closed vessel provided with suitable inlet and outlet passages controlled by valves. In some cases we may have to deliver a large volume of water against but a small head, pressure, or resistance, whilst in others there may be only a small volume of water to be delivered but a great resistance to be pumped against, and it is evident that although similar parts are to be found in pumps for these and other services, their precise construction, arrangement, and relative dimensions must vary with the nature of the work to be performed. We will, however, first consider certain laws and conditions applicable to the working of pumping machinery in general, before proceeding to discuss some typical examples in detail.

Height of Suction and the Energy Required to Raise Water from a Lower to a Higher Level.

It has been not uncommon in the past to meet with instances where those responsible for the erection of a pump have carefully given it an excessive suction lift (in spite of the known difficulties involved), because they suffer

under the delusion that by so doing they are getting a certain amount of work for nothing. Every foot of suction, they will tell you, means so much the less head to pump against; and it is to be feared that victims of the same fallacy are more plentiful than might be expected. A little thought should make it quite clear that a pump is not a creator of power.

The adjoining fig. 1 is a sectional sketch diagram representing a single-acting force pump with the plunger or bucket A moving in the direction indicated by the arrow *a*. When the pump is first started the pressure of the atmosphere acts equally upon both sides of the plunger A, which is therefore (neglecting its weight) in equilibrium. But after the plunger has been reciprocated a few times within its barrel or cylinder, the air is exhausted from the suction pipe B, whose lower open or perforated end is below the surface of the water to be raised. Such water, therefore, flows up the suction pipe, under the influence of the atmospheric pressure on its surface (as indicated by the arrows *b*), and enters the pump barrel or cylinder. Now, the normal pressure of the atmosphere is 14·7 lb. per square inch, which will support a column of water about 33 ft. high. If, therefore, the height C, when the plunger is at the *top* of its stroke, is 23 ft., it follows that of the atmospheric pressure available at the base of the suction column 23/33rds is required to balance the suction column itself, whilst only 10/33rds is available as a balancing force on the under side of the plunger against the constant atmospheric pressure (equal to about 33 ft. water head) on the top of it. Thus, although it is true that the atmosphere raises the water in the suction pipe, it can only do so by neglecting, if we may thus put it, its proper function of balancing the atmospheric pressure on the other side of the plunger. It is simply the old story of "robbing Peter to pay Paul," for every foot of suction lift means an extra foot for the top or delivery side of the plunger to work against.

In every case, then, the work required to raise water by a pump from a lower to a higher level is the same, whatever may be the suction lift. Thus, if a pump is required

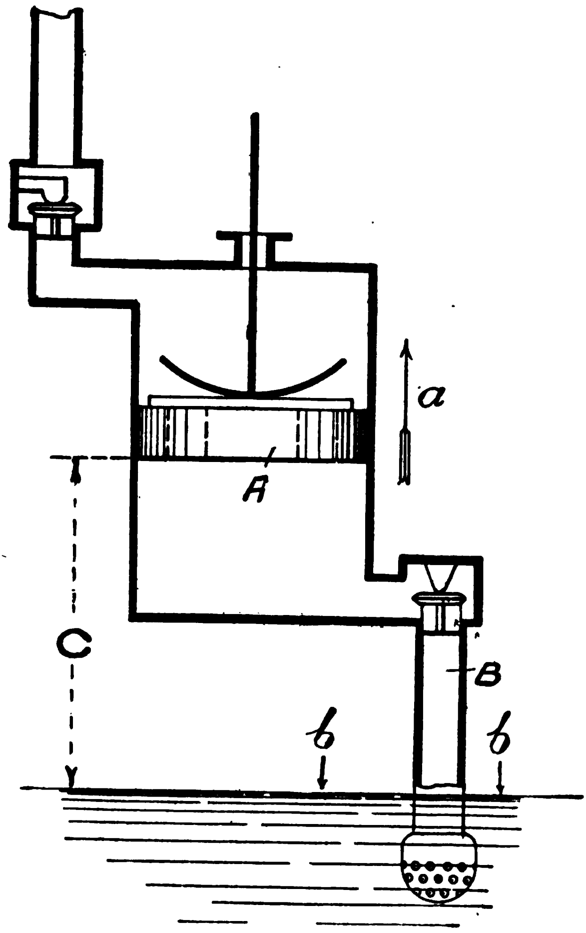


FIG. 1.

to deliver 10,000 gallons (100,000 lb.) of water per hour from a river to a reservoir situated 100 ft. above such river, the amount of work to be performed (exclusive of friction) is:

$$100{,}000 \times 100 = 10{,}000{,}000 \text{ foot pounds per hour.}$$

$$= \frac{10{,}000{,}000}{33000 \times 60} \text{ H.P.}$$

$$= 5{\cdot}05 \text{ H.P.}$$

A long suction lift is, then, of no advantage as regards the expenditure of power necessary to drive the pump. And that is not the end of the matter, for such a lift is a decided disadvantage, in that it operates or tends to operate against the smooth and steady working of the machine. With a long suction lift and a quickly moving plunger there is a difficulty, unless the pipes are excessively large, in making the incoming water keep up with the movement of the plunger within the water cylinder or barrel. Further, the sudden stoppage of the suction column on the reversal of movement of the plunger at each stroke end sets up a decided hammer action. The longer the suction lift the greater the amount of momentum to be taken up; and where, therefore, a heavy suction lift is unavoidable, it is very desirable, and frequently a necessity, to employ a suction chamber, or vacuum chamber, upon the suction pipe to relieve the pump itself of the impact. In a future article we shall refer to the proper disposition of the suction chamber and of other pipe connections and fittings. But as an example of the force of the impact or hammer action in a suction main, let us here consider the case of a single cylinder pump, having a plunger 20 in. diameter, a stroke of 2 ft., a suction pipe 11 in. diameter, with a suction lift of 20 ft., and working at the rate of 70 strokes a minute. As the area of the water cylinder is 314 square inches (or 3·3 times greater than the pipe area), it follows that the velocity of the water through the suction pipe must be 3·3 times greater than the plunger speed, or $70 \times 2 \times 3{\cdot}3 = 462$ ft. per minute, or 7·7 ft. per second. Now, the energy in foot pounds of the moving

suction column, like the kinetic energy of any other moving body, is expressed by the well-known formula:

$$\frac{W\ v^2}{2g}$$

Where W=weight of the body in pounds,
v=velocity in feet per second,
g=accelerating force of gravity, say 32·2 ft. per second.

Taking the quantity of the water (in cubic feet) in the suction column simply as the product of the pipe area and the height of suction, we have (neglecting any horizontal distances):

$$20 \times \frac{95}{144} = 13{\cdot}2 \text{ cubic feet.}$$
$$= 13{\cdot}2 \times 6{\cdot}23 \text{ gallons.}$$
$$= 13{\cdot}2 \times 6{\cdot}23 \times 10 \text{ lb.}$$
$$= 822 \text{ lb.}$$

Therefore, by substituting the figures for the symbolic letters in the formula, we have as the energy of our moving suction column:

$$\frac{822 \times 7{\cdot}7 \times 7{\cdot}7}{2 \times 32{\cdot}2} = 756{\cdot}77 \text{ foot pounds}$$

With these figures before us, it is evident that unless some such provision as a vacuum or suction chamber is provided for the absorption of the energy of the suction column (as a flywheel absorbs the surplus energy of an engine), it will set up a decided hammer action and impose a great shock on the pump at the end of each stroke when the flow of water is suddenly arrested.

Limit of Suction Lift.

As the pressure at the base of a column of water about 2·3 ft. high is just 1 lb. per square inch, it follows that under the normal atmospheric pressure of 29·9 in. of mercury, or 14·7 lb. per square inch, the maximum

theoretical height to which water can be raised by the atmosphere is:

$$2{\cdot}3 \times 14{\cdot}7 = 33{\cdot}8 \text{ ft.}$$

But for such a suction lift a perfect vacuum in the pump chamber and throughout the suction main is an essential condition. Moreover, even if a perfect vacuum were obtainable the pump would not work with such a suction lift, as there would be no energy available to overcome the frictional resistance to the flow of water, and to provide that excess of force, beyond what is due to merely balance the column, necessary to keep the supply to the water chamber equal to the rate of displacement therefrom. Satisfactory pumping is sometimes effected under a suction lift of 26 ft., but as a general rule it is advisable not to exceed 20 ft.

It frequently happens that in addition to a heavy vertical lift, a pump must be so fixed in relation to the source of water that a great horizontal length of suction piping is necessitated. In such a case it must be remembered that the atmospheric pressure acting on the surface of the water to be raised will have to perform the additional work of driving such water through the long horizontal portions of the suction main, and will therefore be that much the less available for balancing the atmospheric pressure acting on the delivery side of the plunger.

When pumping against heavy pressures such as are employed in many hydraulic services, it is advisable to work without suction lift, making the water to flow into the pump chambers under a sufficient head to ensure that they shall at all times be fully charged. In a power pump working under a pressure of, say, a ton or more per square inch, it is particularly necessary that the incoming water shall never lag behind the displacing action of the plungers or rams; if it does so, the resulting concussion will suggest the working of a steam hammer rather than a steam pump.

When pumping hot liquid, it is also necessary to avoid suction lift, and thus ensure that the pressure in the upper part of the rising or suction main shall not fall below

atmosphere. At atmospheric pressure the boiling point of water is 212 deg. Fah., but, as is well known, a fall of pressure is accompanied by a fall in the boiling point, the latter being as low as 162 deg. Fah. under a pressure of 5 lb. absolute, or about $\frac{1}{3}$ of an atmosphere. With water at a high temperature, and with a pump so fixed as to have a heavy suction lift, the machine may be run all day without raising a drop of water; the displacing action of the pump plungers simply bring about an evaporation of the water in the suction main, and thus steam or vapour will be delivered instead of water.

The Effect of Air in a Pump Chamber.

It is unnecessary to dwell upon the importance of preventing access of air to a pump suction main and connections, for it is obvious that if air does get in, the maximum amount of water cannot rise through the suction main, and the pump will consequently fall short of its proper delivery. But in certain types of pumps in which, for reasons that will appear later, the capacity of the pump chamber is considerably in excess of the displacement of each pump stroke, it may happen that the air in such chamber will prevent the admission of water from the suction main. A reference to the adjoining sectional sketch diagram, fig. 2, will perhaps assist in making the matter clear. The pump barrel A is fixed centrally within the large double-acting pump chamber B, having suction valves *a a* at bottom and delivery valves *b b* at top. Let us assume that after the pump has been working for some time delivering water to a tank fixed at a considerable height above the pump chamber, it has become necessary to make an internal examination of the chamber. For that purpose the valve C in the delivery pipe D is closed to prevent return of water, and the necessary covers of the chamber are then opened. At the conclusion of the inspection, and of the repairs, if any, the chamber covers are securely jointed, the valve C reopened, and the pump started again. It will probably be found, however, that no water will be delivered, because the full pressure of

water in the rising or discharge main or pipe D acting upon the valves *b* will keep them closed against the air, with which, of course, the chamber became charged when its covers were opened. It follows that the air will not be dislodged, but simply compressed by the reciprocatory

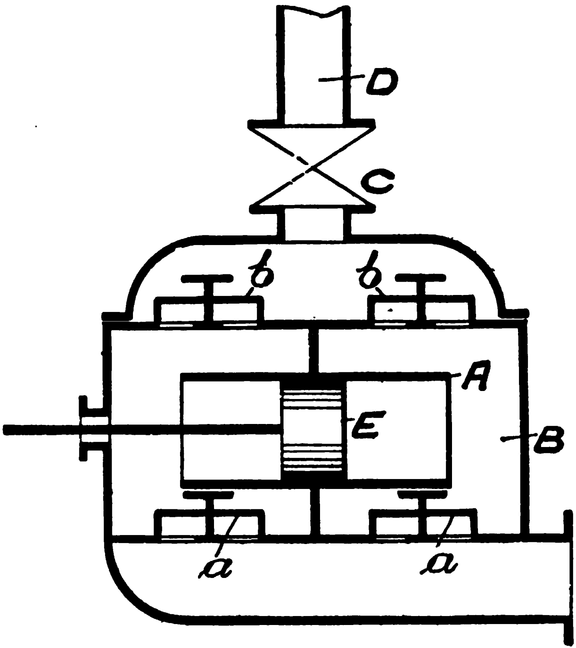


FIG. 2.

motion of the plunger E. In all such cases provision should be made (in some such manner as will be explained in a future article) for the working of the pump with little or no head or pressure upon the delivery valves *bb* until all the air is expelled from the chamber.

CHAPTER II.

Piston Speeds.

The speed at which a pump piston or plunger may be safely driven varies with the conditions under which it is working. For obvious reasons, no user will want to purchase a larger pump than is absolutely necessary, and as the capacity of the machine (measured by the amount of water it will deliver in a stated time) is directly proportional with the speed, it follows that a pump which one maker will offer as having a capacity of, say, 5,000 gallons per hour, may be advertised by a less cautious firm as being capable of delivering 6,000 gallons in the same time. The difference is simply that in the latter case the pump must be run 20 per cent faster than in the former, and, of course, the steam consumption, or the power required to drive it, will be increased by a like amount.

Where the total resistance to be pumped against does not exceed about 370 ft. water head, or a pressure of 160 lb. per square inch, a piston speed of 60 ft. per minute is good practice for the steady daily working of a steam pump. This speed is frequently exceeded, and, under certain conditions, with advantage. Many long-stroke pumps are working with a plunger speed as high as 200 ft. per minute, and large pumping engines have been built and are said to have done good service against a low water head or pressures when running at 600 ft. per minute plunger speed. On the other hand, with hydraulic-pressure pumps, a speed of 20 ft. per minute may, in some cases, be quite fast enough, or even too fast, for satisfactory working.

With a quick-running pump it is very essential to provide large valve areas and water passages, and especially so on the suction side, to ensure that the pump barrels shall be completely filled with water at each stroke.

It must also be remembered that the frictional resistance increases very rapidly when the water is propelled through the pipes at a high speed. At velocities up to about 3 ft.

per second, or 180 ft. per minute, the frictional resistance is simply proportional to the speed, but at higher velocities the friction may increase as much as the square of the speed, or even at a greater ratio. Thus it is calculated that when water is forced through a 4 in. iron pipe at the rate of $1\frac{1}{4}$ ft. per second, or 75 ft. per minute, the loss in pressure due to friction is less than 1/10th lb. per square inch per 100 ft. length of pipe, whereas at a velocity of 19 ft. per second, or 1,140 ft. per minute, the frictional loss will rise to rather more than 14 lb. per square inch for every 100 ft. length.

Pipe Areas.

When determining the dimensions of the pipe connections of a pump, regard must be had to the question of friction just referred to, and on the suction side we must also consider the laws relating to falling bodies. It is especially essential to consider the relationship between the velocity of the moving body and the height from which it has fallen to acquire such velocity. The said relationship is expressed by the formula—

$$h = \frac{v^2}{2g}$$

where v = velocity in speed per second, and g = accelerating force of gravity, say 32·2.

It follows, therefore, that the velocity varies as the square root of the head, and that the theoretical velocity of water under any given head can be thus expressed—

$$\text{Velocity in feet per second} = \sqrt{\text{head in feet} \times 2g}.$$

As an example, let us refer to the sketch diagram, fig. 3. A represents a closed vessel in which we have a vacuum of, say, 26 in. of mercury, which is equivalent to a pressure represented by 28·7 ft. of water head or $12\frac{1}{2}$ lb. per square inch below atmosphere, or 2·2 lb. absolute pressure. Let the vessel A be connected by a pipe, as shown, with the tank B open to atmosphere, and containing water up to the level of the base of A. It will be readily understood that the unbalanced pressure on the under or pipe side of

the closed valve C will be that due to a water head of 28·7 ft., or $12\frac{1}{2}$ lb. per square inch, corresponding with the vacuum within the vessel A. On opening the valve C the water will flow through it into the vessel A, and if we assume that the vacuum is maintained (the water being withdrawn immediately it enters A), and that the level in the tank B is kept constant by an inflowing stream, then the effect, so far as the velocity through the valve is concerned, will be the same as though the water fell through

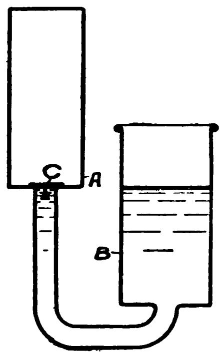


FIG. 3.

a pipe from a tank having an internal pressure equal only to the pressure within the vessel A, but situated at a height of 28·7 ft. above the valve. The velocity will therefore be—

$$\sqrt{28{\cdot}7 \times 2g} = 43 \text{ ft. per second.}$$

But in a case where the suction valve (or the valve C at fig. 3) is 20 ft. above the level of the water to be raised, then it will be seen that the unbalanced pressure under the valve (with a vacuum of 26 in. of mercury in A) will be only $28{\cdot}7 - 20 = 8{\cdot}7$ ft. The velocity will thus fall to—

$$\sqrt{8{\cdot}7 \times 2g} = 23\tfrac{1}{2} \text{ ft. per second,}$$

and this will be the theoretical velocity; the actual velocity through the pipes and connections of a pump will be considerably reduced by the frictional resistance. In

actual practice, with pumps having piston speeds from 60 ft. to 100 ft. per minute, it is usual to employ suction pipes of such dimensions that the velocity of water through the same shall not exceed from about 270 ft. to 360 ft. per minute, or from 4½ ft. to 6 ft. per second. Somewhat smaller pipes may be employed on the delivery side.

As an example, we give in the table below the pipe sizes recommended by the Worthington Pump Company Limited, for use with their well-known regular pattern pumps, which are designed for general services where the steam and water pressure does not exceed 160 lb. per square inch. To indicate how much greater must be the

SIZES OF PIPES FOR WORTHINGTON PUMPS.

Dimensions of pump.			Strokes made by each plunger per minute.	Gallons per minute delivered by pump.	Pipes.			
Diameter of steam cylinders in inches.	Diameter of water cylinders in inches.	Length of stroke in inches.			Steam.	Exhaust.	Suction.	Delivery.
4½	2¾	4	100 to 200	17 to 34	½	¾	2	2
6	4	6	100 to 150	54 to 80	1	1½	3	3
9	5¼	10	75 to 125	115 to 190	2	2½	4	3
10	6	10	75 to 125	150 to 250	2	2½	5	4
12	8½	10	75 to 125	315 to 500	2½	3	6	5
14	10½	10	75 to 125	440 to 740	2½	3	8	7
16	12	10	75 to 125	610 to 1,000	2½	3	10	8
18½	14	10	75 to 125	820 to 1,380	3	3½	12	10

area of the pipes employed for the conveyance of a liquid than that of the pipes conveying a gaseous body, we have included in the list the sizes of steam-supply and exhaust pipes recommended by the makers. In every case it will

be seen that although the area of the steam cylinders exceeds that of the water cylinders (or pump barrels), the pipes for the latter use are much larger than those employed at the steam end of the pump.

In calculating the capacity of Worthington pumps (or the quantity of water they will deliver in a given time), it must be remembered that the Worthington pump has *two* double-acting water plungers; its capacity, therefore, is twice that of any ordinary or simplex double-acting pump of same size, but having only one steam and one water cylinder, and four times as large as a single-acting pump.

Let us consider the first pump given in the list, when running at such a speed as to deliver 34 gallons per minute. As each plunger must then make 200 strokes per minute, and as the length of stroke is 4 in., or $\frac{1}{3}$ ft., it follows that the piston speed must be—

$$\frac{200}{3} = 66 \cdot 7 \text{ ft. per minute.}$$

Now, the area of each $2\frac{3}{4}$ in. water plunger is 5·93 square inches, and the area of the 2 in. suction pipe is 3·14 square inches. But though the pump has two water plungers, it has but one suction pipe, and thus the ratio between the combined areas of the water plungers and the area of the suction pipe will be $2 \times 5 \cdot 93 : 3 \cdot 14$, or $3 \cdot 77 : 1$. Therefore, as the piston speed or plunger speed is 66·7 per minute, it follows that if the water cylinders are to be kept fully charged, the velocity of the water through the suction pipe must be not less than $66 \cdot 7 \times 3 \cdot 77 = 251 \cdot 5$ ft. per minute, or nearly $4\frac{1}{4}$ ft. per second.

In the same manner it will be found that with the fourth pump on the list (the $10 \times 6 \times 10$), the velocity of the water works out at about $5\frac{1}{4}$ ft. per second, and with the last pump on list at about 6 ft. per second.

It should be noted that as the stated piston speeds represent the average in the given unit of time (one minute), so also the calculated figures will give the average, and not the maximum, velocities of the water through the suction pipe in each case.

AIR VESSELS.

The greater the variation between the maximum and the minimum speed of a pump throughout its stroke, the greater the necessity for an air vessel on the delivery side to equalise the flow of water. In the direct-acting pump the speed is fairly uniform throughout each stroke, though a slight pause occurs at each reversal of the motion. But with a crank and flywheel pump, in which the reciprocatory motion of the plunger is obtained from the rotatory movement of the crank, the speed of the plunger is ever varying throughout its stroke A single-acting crank and flywheel pump is the type with which an air vessel is most needed, whereas with a duplex direct-acting pump an air vessel may, in many cases, be entirely dispensed with, and the smooth, quiet, and steady working of the pump still be maintained.

In his "Mechanics of Pumping Machinery," Weisbach, considering the size of an air chamber for a manual fire pump or fire engine, says that its capacity should be at least eight times that of the pump cylinder. In simplex or single-cylinder single-acting pumps, it is no doubt well to keep to some such ratio, large as it may seem to those accustomed only to the much smaller air vessels, which (if necessary at all) are found to be accompanied by a uniform water delivery in quite large duplex double-acting pumps.

A pump may be provided with an air vessel or chamber of more or less fanciful design and got up in burnished brass or copper, but whether it serves any useful purpose will depend upon what it contains. When the pump was first started the vessel was, of course, charged with the substance which gives its name, but after the pump has been working some time it is more than probable that all the air has been gradually absorbed, or withdrawn by the outflowing stream, and the vessel become completely waterlogged. To ensure that the air vessel of a pump is really a vessel containing air it should be fitted with a gauge glass, and stop valves should be provided to permit of the ready withdrawal of the accumulated water and re-charging with air.

For the automatic charging of an air vessel with the ordinary working of the pump to which it is attached a snifting valve is sometimes provided to admit air into the pump chamber or barrel on each suction stroke. It is better practice to arrange an automatic air-charging device separate from the pump barrels, and so avoid the reduction of the capacity of the pump which results from the former system.

Fig. 4 is a sectional view of one type of air vessel suitable for direct attachment to a pump chamber. The lower end of the delivery pipe or discharge main A dips or passes into the interior of the vessel, as shown, and hence is termed the "dip pipe." The lower end, or water inlet end, of the

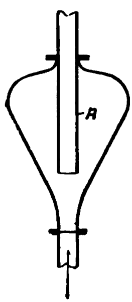


Fig. 4

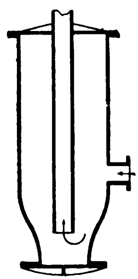


Fig. 5.

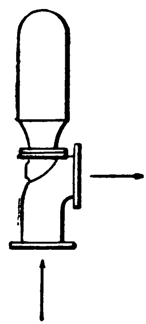


Fig. 6.

vessel is provided with a narrow neck, as shown, whilst the upper part of the vessel is splayed, to give the advantage of a large water surface in contact with the air, such surface being less liable to disturbance by the flow of the water.

To give a greater opportunity for the separation and lodgment in the vessel of any air brought in by the water, an arrangement such as is shown at fig. 5 may be adopted. In this case the incoming water must, as will be observed, fall towards the bottom of the vessel in order to enter the dip pipe, and during such descent the air has a good chance of rising to the upper part of the vessel. For the same

reason air vessels are sometimes provided with double dip pipes, the incoming water being passed through one pipe, which discharges at the top of the vessel, whilst the outgoing water is discharged through the other pipe, which has its open end near the bottom of the vessel. The pipes enter the vessel centrally, but are suitably curved to enable them to clear each other in its interior.

Fig. 6 is an illustration of a convenient type of air vessel of a small capacity. It consists of a plain, narrow-necked chamber, closed at one end and bolted to a bend, which is itself mounted directly upon the top of the pump chamber.

CHAPTER III.

Arrangement of Pipes and Connections.

Every pump maker knows how frequently it happens that the working of one of his machines, in every respect suitable for the service required of it, is much impaired or rendered altogether impossible through a bad arrangement of pipes and connections.

With many users the provision of an extreme suction lift (to obtain the imaginary advantage referred to in our first chapter) is the chief error to be encountered. But though with better informed people an excessive suction lift will not be adopted, other errors which result in considerable trouble and inconvenience are not always avoided.

Makers of wide experience are careful to intimate in their catalogues that when a pump draws from a supply of water beneath it, no part of the suction main should rise above the level of the pump-chamber inlet. Or, to put it in other words, the entire length of the pipe or main should fall from the pump to the water supply. If this warning be acted upon, then, in a case such as illustrated at fig. 7, in which a pump chamber A (shown in end elevation), must of necessity be placed on the side of a wall B, away from the sump or source of the water supply C, the erector will

be careful to avoid carrying his suction pipe over the top of the wall (in the manner indicated by the dotted lines), even though the height of the wall above C is well within a moderate suction lift. But if, disregarding the warning, he *does* adopt the dotted line arrangement for his suction pipes, he may save himself trouble for the time being, but

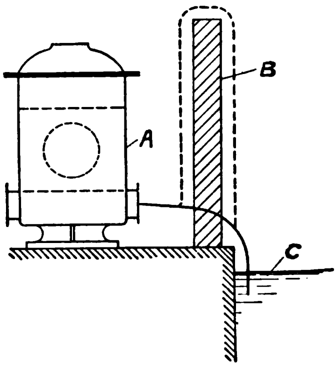


FIG. 7

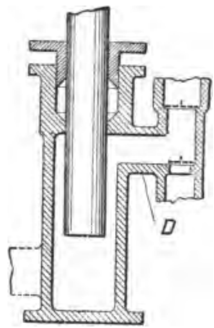


FIG. 8.

there will be plenty of it subsequently from the return or syphon action on each reversal of plunger movement. The pipe should in this and similar cases be carried *through* the wall, as shown by the full line in the figure.

Pump designers themselves are not, however, always careful to so arrange the delivery valves and outlet connections on their pump chambers as to avoid the lodgment of air within such chambers. Fig. 8 is an illustration of a single-acting ram pump, in which the branch or connection D, containing the suction and delivery valves, is arranged at the top of the chamber. As the delivery valve is, in this arrangement, above the chamber proper, any air coming in with the water through the suction valve will readily flow out on the down stroke of the ram, through the delivery valve. But if, instead of the valve branch being arranged as above described, it is disposed in a lower position on the pump chamber, as indicated by dotted lines on the left

hand of the figure, any air brought into the chamber may become lodged in the upper part of it, thereby limiting the capacity of the pump.

Position of the Vacuum Chamber.

We have previously considered the necessity for the provision of a vacuum chamber, or suction chamber, to absorb the energy of a long suction column. But in order that it may properly perform such a function the vacuum chamber must be placed in a favourable position. At fig. 9 three positions for the suction chamber are indicated by dotted lines. The position A at right angles to the flow of water is a bad one, as the impact would be imposed on the pump

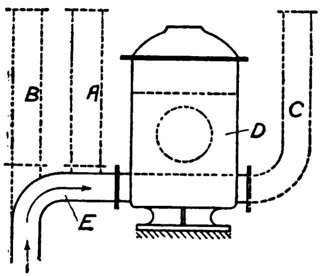


Fig. 9.

before the water had time to alter its course for the flow into the vacuum chamber. With the position B the water flowing up the vertical suction main can ascend into the vacuum chamber without any alteration of its course. The position C is a good one in the case of a long length of horizontal suction main. As will be seen, the vacuum chamber is arranged on the side of the pump chamber D, opposite to the suction main E, so that the water can flow right through the lower part of the pump (beneath the suction valves), and by an easy bend pass into the vacuum chamber.

Strainer, Foot Valve, and Charging Connections for Suction Pipe.

The general practice, in this country at any rate, is to fix the strainer (when such an attachment is necessary) upon the bottom of the suction pipe. The advantage of this position is that solid matter such as the strainer arrests is kept clear of the foot valve. It may be noted that in making a strainer care should be taken to have the combined areas of the strainer holes not less than two or three times in excess of the suction pipe area, in order to allow a sufficient inflow even though a number of holes may be blocked.

But the difficulty of getting at the strainer to free it from obstruction when it is fixed on the end of the suction pipe has led to the adoption of the arrangement indicated to the left hand in fig. 10. The strainer S is arranged, as is

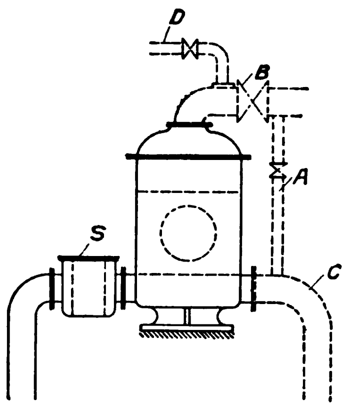


Fig. 10.

shown, adjacent to the pump chamber, and thus it can be very readily inspected and cleared as may be required. The strainer should comprise two main parts, viz., the strainer proper consisting of a wire basket or perforated diaphragms, and a box or casing enclosing the same, but permitting of ready insertion or withdrawal.

Foot valves, like check or retaining valves generally, are made with the moving parts, or valves proper, arranged in various ways. The types to avoid are those in which wing like or other projections are formed on the valve itself to guide its movement, for such a valve is very liable to become clogged, and so stuck up off its seating. The valves should always be of the flat or disc type, having nothing below the surface which rests upon the fixed seating, and thereby forms the joint. A multiple type foot valve containing a number of small metal disc valves with leather faces, arranged in the one chamber or casing, has been found advantageous.

The dotted lines at fig. 10 indicate a convenient arrangement of pipes and connections for charging the pump chamber and suction main with water from the delivery main before restarting the pump after a stoppage. For after a stoppage it may be, as explained in our first chapter, that the pump will not lift the water because the motion of the plungers simply compresses the air instead of completely displacing or dislodging it. But on opening the valve in the pipe A, fig. 10, which connects the delivery main (beyond the check or retaining valve B) with the suction pipe or main C, the latter and also the pump itself can be readily charged with water. The waste delivery pipe D, when its valve is opened, permits of the escape of the air displaced from the pump chamber. The pipe D may with advantage be left open during the first few strokes of the plungers, but it should, of course, be closed when the pump has fairly caught its water.

Bell-mouth for Suction Pipe, and Nozzle for Delivery Pipe.

Just as a stream of water issuing from an orifice is contracted to an area less than the area of the orifice (such contraction being termed *vena contracta*), so a contraction of the stream occurs when water flows into a pipe. To diminish the effect of such contraction and to facilitate the inflow of water, the lower end of a large suction pipe is sometimes rounded or is provided with a bell-shaped mouth. And to give a steady and full bore or solid dis-

charge from the delivery pipe the outlet end of such pipe is sometimes provided with a conical or curved discharge piece or nozzle. A "fireman's nozzle" is a good example.

CHAPTER IV.

Plungers and Plunger Packing.

Pump plungers may be either of the "externally-packed" or the "internally-packed" type. In general, externally-packed plungers are styled "rams," and pumps fitted with the same are frequently known as ram pumps. In like manner pumps fitted with internally-packed plungers are sometimes known as plunger pumps or piston-pattern pumps.

Each type has inherent advantages and disadvantages, which we will briefly consider. The sketch diagrams, figs. 11 and 12, representing a ram and a piston-pattern pump respectively, will serve to keep the two types before us during such consideration.

In the ram pump, fig. 11, there is simply a stuffing box to be kept packed, whereas in the piston-pattern pump,

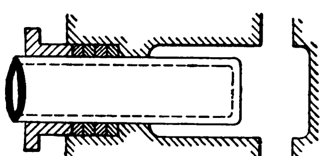


Fig. 11.

fig. 12, we have both a stuffing box and the internal plunger or piston to attend to. A stuffing box can be very readily packed, but it is frequently a difficult and troublesome business to pack a plunger, for the end cover joint must be broken and subsequently re-made, and if the plunger, rod, and plunger cap are made of iron, the three parts will probably be so rusted together that the task of removing

the cap to permit of the insertion of the packing may involve the expenditure of much time and temper. A man who has had such an experience will be a strong advocate for the adoption of brass plungers and rods.

A ram pump can be used for dealing with gritty water, whereas, with a plunger or piston-pattern pump on such a service, waterways are speedily cut along the plunger and the barrel, with the result that a quantity of the liquid is simply churned back and forth in the barrel instead of being displaced from the same.

Leakage of water through the gland of a ram pump is at once observed, but the leakage past a plunger cannot, of course, be seen. In the latter case, the leakage is revealed only when it has become so extensive as to seriously diminish the output or delivery of the machine.

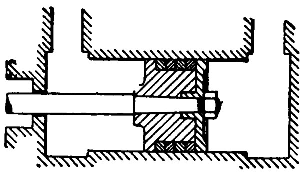


FIG. 12.

Turning now to the other side of the question, we have to remember a ram is ordinarily single acting, delivering water only on every second stroke. A plunger or piston can be double acting, so that when running at the same speed it will deliver double the quantity of water discharged in an equal time by a ram of the same diameter and length of stroke. A ram-pattern pump can be made double acting, but two rams, or a single ram with divided chamber, must be employed. It is obvious, therefore, that in the matter of compactness and simplicity of outline the advantage is with the piston pattern.

On many services it is of prime importance that the capacity of the water cylinder shall be but little in excess of the displacement effected on each pump stroke, or, in other words, that the amount of air space or clearance in

the water cylinder shall be reduced to a minimum. For such services ram pumps are quite unsuited.

Unless the glands of ram pumps are most carefully packed, a very large proportion of the work put into the pump will be required simply to overcome the friction between the packing and the surface of the ram. It is a very easy matter to pull up a pump by tightly screwing down the gland, and, unless otherwise directed, an attendant, in his anxiety to prevent leakage through the stuffing box, may put in so much packing that he must perforce vigorously ply hammer and drift before the studs can be made to project through the face of the gland; the latter is then screwed down till the perspiration rolls from the forehead of the honest but thoughtless workman.

The packing that may give every satisfaction for the glands of steam piston rods is generally quite unfitted for the glands of pump rams and water pistons or pump rods.

At the water end of the steam pump the gland packing should be soft and well greased throughout. For the rams of pumps for hydraulic presses and other high-pressure services, self-acting leather packings of the U or hat types may be employed with cold liquids, but owing to their annular form they cannot be so readily inserted as hemp or like packing, and hence the latter is more generally adopted.

With a piston or plunger, a self-acting packing of the well-known cup leather type can be fitted without great difficulty. The pump barrel should, however, be invariably brass lined, for self-acting packings are very rapidly destroyed unless the surfaces against which they work are quite smooth and free from corrosion.

Pump water pistons or plungers may be packed with soft hemp or similar packing in the manner indicated at fig. 12. They are sometimes also packed with rings made of a non-corrosive metal. Plungers without any packing whatever give every satisfaction on services where there is but a moderate head or pressure (not exceeding about 150 lb. per square inch) to be pumped against. In such cases, however, the plunger should be made somewhat longer than an ordinary packed piston to provide ample

wearing surface. The slight leakage of water past a plunger of this description is more than compensated by its freedom from the friction involved in the use of a packed plunger.

VALVES.

The provision of multiple valves on pump chambers is now a universal practice. In the well-known flat-disc form they appear to have been first generally adopted in America upwards of 50 years ago, in connection with boiler-feeding and bilge pumps for steamships. "All pumps for this service," says an American writer, "were formerly provided with only one valve to each end of the pump chamber. The valve was flat and oblong, covering a port like that of a steam chest, and rocked open and shut on a half hinge on one side, governed by a flat spring of hard brass. A small obstruction under the heel of this valve would throw

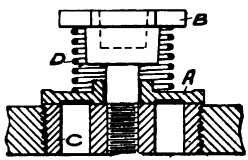


FIG 13.

that end of the pump out of commission. It was quite common to find pieces of wood, scraps of leather and rubber, and sometimes a fireman's cap and overalls in the valve chambers."

"The multiple valve," proceeds the same writer, "scored a great success, as the valve plates acted as a coarse strainer, and all the débris and rubbish that came up from the bottom of the ship stopped in the large suction chamber, and could be removed from time to time as the pump showed signs of strangulation."

Fig. 13 is a sectional elevation of the metal disc type of valve, several of which can be fitted, when the multiple valve system is adopted, to control the suction and delivery ports or apertures of a pump chamber; fig. 14 is a plan

of the valve seating. The valve itself consists of a metal disc A, or a rubber disc may be employed for cold water. The guard B is screwed into the valve seat C, and the latter is itself screwed into position in the pump chamber. The spring D retains the valve on its seating. As there is no projection of the valve below its seating, it cannot become "stuck up" therein.

As already stated, a number of valves of the type illustrated can be arranged on the one pump chamber, but in pumps for working against heavy pressures, though the disc type of valve can still be advantageously employed,

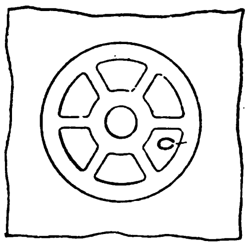


FIG. 14.

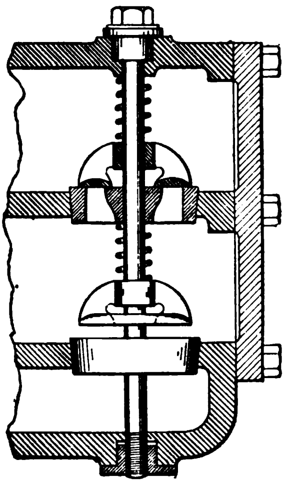
each valve (or group of valves) should be disposed in a separate pot or valve chamber. The detachable head or cover of each valve pot or chamber can be provided with a hollow projection or socket to serve as the valve guard and stop, a corresponding central projecting stem being formed upon the upper side of the valve to enter the said socket.

When rubber discs are used in place of the metal discs, as A, fig. 13, a metal plate or washer should be interposed between the back of the valve and the metal spring.

Fig. 15 represents, in sectional elevation, a suction and a delivery valve arranged on one central stem. The valves are of the type sometimes known as "double-way," because the water can flow out under both the inner and outer edges of the ring-like bearing piece of the valve when the latter is raised from its seating, whereas in the valve shown at fig. 13 the water has but one annular space to

flow through. A plan of one of the double-way valves is shown at fig. 16. Tubes or hollow buffers of indiarubber are sometimes employed instead of the valve springs; and we here note that in all cases it is advantageous to employ

FIG. 15.



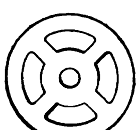


FIG. 16.

springs or buffers of such a normal length that the valves have a certain amount of free lift before commencing to compress the springs.

The old form of flap or clack valves is still frequently adopted, especially for services where the water is gritty

or contains solid matter. The valve in its simplest form is readily made from a pad of leather which is secured on one side to the valve seat, such secured side serving as the hinge. The leather is topped with a metal plate or disc, the two parts being fastened together by means of copper rivets. Though suitable for low lifts and slow speeds, the objection to this type of valve is the necessity for allowing them a large movement in order to obtain a free waterway; the valves are, therefore, comparatively slow in closing, with the result that the "slip," or return of water through them, is considerable. Large valves

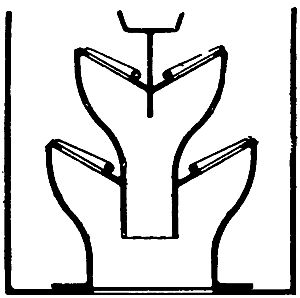


FIG. 17.

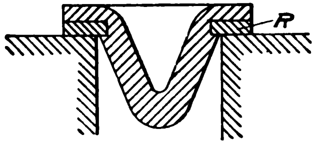
provided with several clacks or flaps, are termed "multiple-clack" valves. They are made in many forms; one type is shown in outline at fig. 17.

A very simple type of valve suitable for small slow-speed pumps is shown in section at fig. 18. No guide is needed with such a valve, as it will always right itself in seating. The valve itself, or the "valve fall" as it is sometimes termed, consists of a cast-iron hollow cone, as illustrated at fig. 19, which has simply to be faced under the head and necked for the reception of the rubber ring R; the latter is sprung into position on the fall.

Perpendicular lift valves can be provided with two or more seats, either in the same plane or one above the other, and for large pumps such "double-beat," "treble-beat,"

or "multiple-beat" valves in various forms are commonly employed. Fig. 20 illustrates a valve in which the "beats," or parts which come into contact with the seat, are arranged

FIG. 18.



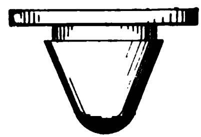


FIG. 19.

in the one horizontal plane. Such a valve may be termed a "quadruple-way valve," seeing that the water will escape through both the inner and outer annular spaces formed

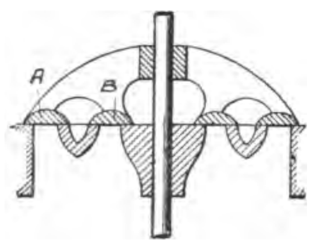


FIG. 20.

between each of the ring-like parts, A and B, when the valve is lifted from its seat.

Many other varieties of multiple-beat valves are designed and produced to suit sundry requirements, fads, and fancies.

Mechanically-operated Valves.

The valves previously considered are opened solely by the water pressure imposed upon them, and it is obvious that they cannot open until *after* the application of the pressure. In practice, no great difficulty is experienced with such valves, even when running at high speeds, if the pump is well designed throughout and adapted to the required service. But in certain pumps of German origin the valves at the water end are, under the Riedler system, positively operated by mechanism worked by a moving part of the pump. The valves are said to "operate with a liberal lift, avoiding any throttling. The mechanism is exceedingly simple. Each valve is closed at the moment the stroke of the piston changes, and this closing is done by means of a spindle projecting into the valve chamber. Near the end of the stroke a very small free lift is allowed to the valve, which can be regulated at will." Mine-pumping engines with the Riedler valve system were first built in 1884, and since that year a large number of such engines have been constructed for various countries, but especially for the deep mines of Bohemia, Silesia, Westphalia, and Belgium.

CHAPTER V.

Boiler Feed Pumps.

The advantage of employing a small steam pumping engine, or donkey pump, for the sole purpose of boiler feeding is now generally recognised by steam users. To meet the demand for such pumps, so many makers have entered the market that a purchaser may well find it difficult to make a wise selection. It has been truly said that "the cheapness of an article depends on more things than price," and in the selection of a boiler feed pump it is of especial importance to bear such a statement in mind.

First and foremost, a boiler feed pump must be absolutely reliable. To that end its working parts must be few in

number and of great simplicity and durability. There must be no risk of the steam controlling valve or any other part becoming "stuck up" or failing in action. The user must be assured that so long as he keeps steam supplied to the pump, so long will it continue to work steadily and uniformly, requiring no frantic manipulations of a starting lever from time to time, or any like assistance from the stoker or attendant.

The water in a boiler should be maintained as nearly as possible at one constant level, and the pump should therefore be capable, when running at a moderate speed, of making the feed keep pace with the maximum rate of evaporation in the boiler.

The surroundings of a feed pump are, more often than otherwise, anything but conducive to the sweet running of machinery of any kind, and hence any elaboration of working parts should be carefully avoided. The feed pump may get a little oil now and again, but it seldom gets any other attention. When a feed pump is in a fair working condition the "slip" should not exceed 5 per cent; in other words, the pump should actually deliver an amount of water equal to 95 per cent of the measured displacement. But if the plungers or plunger packings are allowed to get into a bad condition the slip may be upwards of 25 per cent. In such a case the speed of the pump must, of course, be increased to maintain the water level, resulting in a proportionate increase in steam consumption and in the wear and tear of the working parts.

Ratio Between Steam and Water Pistons.

In the sketch diagram, fig. 21, A and B respectively represent the steam and water cylinders of a pump for feeding the boiler C. Each cylinder is fitted with a piston directly connected by a rod as indicated. If the pistons are of equal area, it is evident that they will be in equilibrium when both steam and water connections are opened to the respective cylinders A and B, for whilst the steam pressure will tend to drive the pistons in the direction indicated by the arrow 1, the water pressure will tend to drive them with equal force in the opposite direction. It is clear, therefore,

that with such a pump the area of the steam piston must be in excess of the area of the water piston or plunger if the water is to be driven into the boiler. The ratio between the two must be such as to provide sufficient force to overcome the friction of the moving parts of the pump, of the fluids in motion, and to effect the performance of the work of displacement. The general practice is to adopt a ratio of about 2 to 1—that is, to employ steam and water pistons of such diameters that the area of the former shall be double the area of the latter. With small feed pumps the ratio is

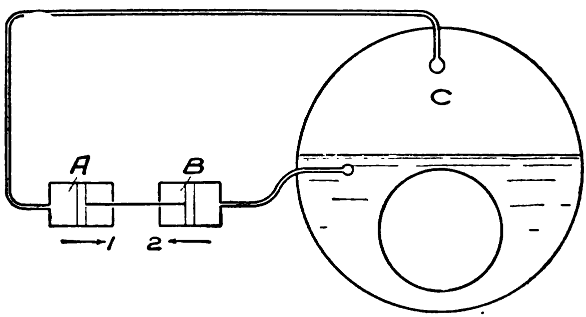


FIG. 21.

somewhat greater, and with larger feed pumps frequently somewhat less, than 2 to 1, for we usually find that the friction is proportionately more in small pumps. As examples, we will here refer to the standard sizes of leading types of feed pumps, of which, by the courtesy and special permission of the makers, illustrations and particulars will be given in the course of these articles.

The ordinary list sizes of the "Favourite" double-acting donkey pumps (illustrated at fig. 22), by Mr. A. G. Mumford, of Culver Street Engineering Works, Colchester, are as follow :—

	in.	in.	in.	in.	in.	in.
Diameter of steam cylinder...	3	$3\frac{3}{4}$	$4\frac{1}{2}$	$5\frac{1}{2}$	6	7
Diameter of water plunger ...	$1\frac{1}{2}$	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	5
Length of stroke	3	4	5	6	6	6

The "Cameron" type double-acting pumping engine (illustrated at fig. 23), by the same maker, is made to the following sizes:—

	in.	in.	in.	in.	in.
Diameter of steam cylinder...	6	7	7½	8	10
Diameter of water plunger ...	3½	4	4½	5	6
Length of stroke	6	7	7	8	9

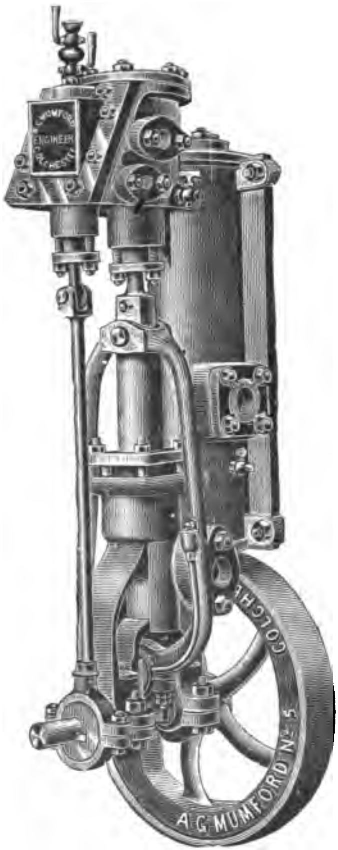


FIG. 22.

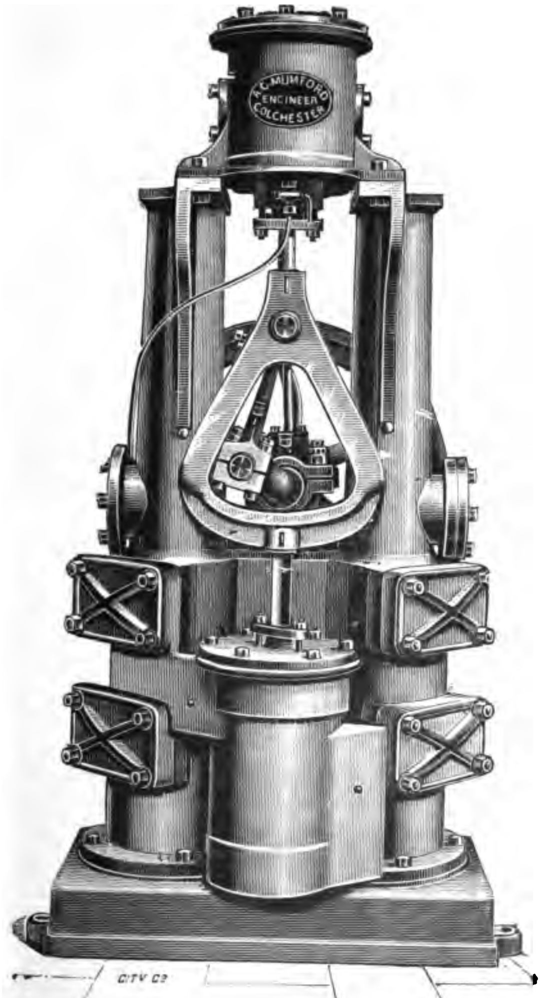


FIG. 23.

The following are some of the sizes adopted by the Pulsometer Engineering Co. Limited, of Nine Elms Ironworks, Reading, in their single-cylinder direct-acting "Deane" boiler feed pumps:—

	in.	in.	in.	in.	in.
Diameter of steam cylinder......	3½	4	5½	6	10
Diameter of water cylinder......	2¼	2½	3¼	4	6
Length of stroke	5	10	10	10	12

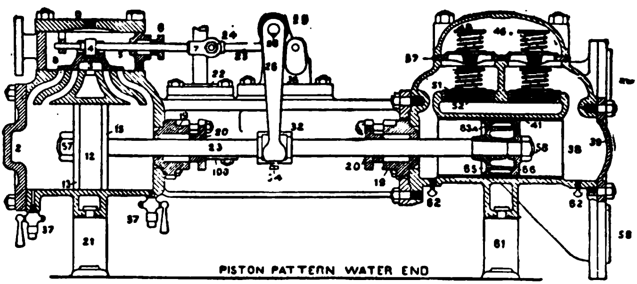


FIG. 24.

Messrs. G. and J. Weir Limited, of Cathcart, Glasgow, adopt the following ratios in their single direct-acting pumps for boiler feeding:—

	in.	in.	in.	in.	in.
Diameter of steam cylinder......	7½	8	9½	10½	12
Diameter of pump cylinder......	5½	6	7	8	9
Length of stroke..................	15	18	21	24	24

The following selection of ratios is from the lists of the Worthington Pump Co. Limited, of 153, Queen Victoria Street, London, E.C. :—

	in.	in.	in.	in.	in.
Diameter of steam cylinder...	2	4½	6	10	12
Diameter of water plunger ...	1⅛	3	4½	7	9¼
Length of stroke	2¾	4	6	10	10

Fig. 24 is a sectional illustration of a Worthington standard type horizontal feed pump, to which we shall again refer in another chapter.

Messrs. W. H. Bailey and Co. Limited, of Albion Works, Salford, adopt the following ratios in their "Davidson" patent boiler feed pumps :—

	in.	in.	in.	in.	in.
Diameter of steam cylinder ..	3½	5½	7	12	14
Diameter of water plunger ...	2	3¼	4	7	8
Length of stroke	4	8	10	12	14

Compound Feed Pumps.

With the high steam pressures now so generally adopted on various services, a demand has arisen for boiler feed pumps having compound steam cylinders. With such pumps a considerable saving of steam is effected; in many cases the high-pressure cylinder can be made of less diameter than the pump barrel.

The following ratios are adopted by the Pulsometer Engineering Co. Limited in their single type compound feed pump :—

Diameter of high-pressure cylinder.	Diameter of low-pressure cylinder.	Diameter of pump barrel.	Length of stroke.
Inches.	Inches.	Inches.	Inches.
4	12	5½	18
5	15	7½	18
7	21	9½	20
9	24	10½	20

The sizes hereunder are selected from the Worthington Pump Co.'s lists of Worthington or duplex type compound pumps suitable for boiler feeding :—

Diameter of high-pressure cylinder.	Diameter of low-pressure cylinder.	Diameter of pump barrel.	Length of stroke.
Inches.	Inches.	Inches.	Inches.
5½	7½	5	6
6	9	7	10
8	12	10½	10
9	14	12	10
10	16	14	10

Messrs. W. H. Bailey's "Davidson" patent compound pump has cylinders proportioned as indicated in the following selections :—

Diameter of high-pressure cylinder.	Diameter of low-pressure cylinder.	Diameter of pump barrel.	Length of stroke.
Inches.	Inches.	Inches.	Inches.
3½	7	3½	8
4	8	4	10
5½	10	5½	12
6	12	6	12
7	14	7	14
9	1	9	18

Steam Consumption in Feed Pumps.

The steam consumption of a feed pump may be stated as the quantity of steam used per horse power per hour, and for general comparative purposes such a statement is very convenient. But it is sometimes more advantageous to have the information expressed in terms giving the quantity of water delivered into the boiler for every pound of steam supplied to the pumps.

Referring again to fig. 21, it will readily be understood that if it were possible for a pump to work with steam and water pistons of equal areas, as therein shown, then, disregarding all condensation losses of every kind, the ratio between the steam consumption and the water delivered into the boiler would be just the ratio between the weight of the steam and the weight of an equal volume of water. Seeing that the steam and water displacements are just equal to each other, it follows that the delivery of a cubic foot of water into the boiler will involve the withdrawal or expenditure of a cubic foot of steam. We thus have a very simple and very convenient ideal standard of reference. The weight of a cubic foot of water is about 62·3 lb., whereas 1 lb. of steam at 100 lb. boiler pressure will occupy a volume of about 3·8 cubic feet. The ideal quantity of water delivered per pound of such steam is thus

$$62{\cdot}3 \times 3{\cdot}8 = 236{\cdot}7 \text{ lbs.}$$

The steam consumption per horse power per hour of our ideal or reference pump, with the aforesaid steam pressure, we can calculate as follows :—

$$\frac{33000 \times 60}{144 \times 100} = 137{\cdot}5 \text{ cubic feet}$$

$$\frac{137{\cdot}5}{3{\cdot}8} = 36{\cdot}1 \text{ lbs. per hour.}$$

Now, although the ideal or reference pump on which the foregoing calculations are based is assumed to work without expansion, yet, even in compound boiler feed pumps using considerable expansion, the results will not equal the figures we have given. In ordinary single-cylinder steam pumps working with little or no expansion, the steam consumption must of necessity be very much higher than our figures, for, as we have already seen, the steam piston must be about double the area of the water plunger, and, in addition, there is the consumption of steam on each pump stroke in the clearance spaces and ports; the said addition may represent more than 15 per cent of the total steam consumption of the pump.

In a paper by Mr. Alexander Borodin on "The Working of Steam Pumps on the Russian South-Western Railways," read before the Institution of Mechanical Engineers in the year 1893, some extremely interesting particulars were given concerning tests conducted on quite a number of pumps of different makes, employed for the water supply at the principal stations. The value of the tests is much enhanced from the fact that the pumps were all tried in their ordinary working condition, without any special preparation with a view to the attainment of good results. We may again have occasion to refer to these trials, but for the present will simply note that the steam consumption ranged from 50 lb. per horse power per hour in a Worthington compound pump to the enormous quantity of 855 lb. per horse power per hour in a small pump made by another maker. The latter pump was not tested by the author of the paper referred to, but full particulars are given of the figures obtained at the test.

"Steam pump manufacturers," said an American writer, in the year 1897, "claim to run on as low as 75 lbs. of water (steam) per indicated horse power per hour, but it is not uncommon to find it double that amount." We may add that compound boiler feed pumps can now be obtained with which the steam consumption is given by the makers as "less than 50 lb. per duty horse power per hour."

As is well known, the injector is an extremely uneconomical appliance for pumping water. An ordinary injector will use about four times as much steam as an ordinary pump doing the same work; but there will be some compensation in the warming of the water.

CHAPTER VI.

Types of Boiler Feed Pumps.

In selecting a feed pump a buyer has first of all to decide between crank and flywheel and non-flywheel pumps. Non-flywheel pumps are frequently called "direct-acting," but inasmuch as many flywheel pumps have the steam and water pistons directly connected, the term is not sufficiently distinctive.

Each type has its advocates, from whom information may be obtained concerning the advantages of the one and the disadvantages of the other.

Flywheel Pumps.

For crank and flywheel pumps the chief claim made is that they are more economical than non-flywheel pumps. Now, while it is perfectly true that many flywheel pumps are working with greater economy than non-flywheel pumps, it is equally true that steam pumping engines of the latter type can be and are made to equal some of the most economical flywheel pumping engines.

But the pumps or pumping engines (for we are now considering pumps having their own steam or motor cylinders) employed for boiler feeding are generally of small dimensions. A boiler supplying a hundred horse power engine can be fed by a pump having a steam supply pipe only three-eighths of an inch in diameter. With such small pumps and running at the slow speeds necessary if the water in the boiler is to be maintained, as it should be, at one constant level, it is obvious that the momentum of the flywheel must be exceedingly moderate, and therefore that the amount of "cut-off" (or expansive working) permissible can be very little, for the work is, of course, a constant quantity.

In boiler feed pumps having the slide valve or steam controlling valve operated from a rotating crank shaft receiving its motion from the reciprocating piston rod, it is, of course, necessary to provide a flywheel to carry the crank over the "dead centres." It is such carrying over that constitutes the chief function of the flywheel. When the pump is running at full speed a slight saving of steam may be effected by expansive working.

A great objection, however, to crank and flywheel pumps is to be found in the difficulty experienced in running them at slow speeds. As an illustration, it may here be mentioned that some of the hydraulic lifts or elevators of the Eiffel Tower were originally worked by flywheel pumps. After the experience gained at the Exposition of 1889 the authorities decided to increase the elevator capacity for the

1900 Exposition, and although they at first only contemplated additions to the existing pumping plant, they finally decided to take out all the flywheel pumping engines, and do the whole elevator service for the tower with non-flywheel pumps. It was found that flywheel pumps were always noisy in action, and that they were continually stopping when it was attempted to run them at such a speed as was sufficient to maintain the amount of water required at certain periods. No such difficulty was experienced with the non-flywheel pumps.

Non-flywheel Pumps.

Non-flywheel pumps may be divided into two main classes, viz., single and duplex. Single non-flywheel pumps are usually associated with a steam operated slide valve, which, as frequently arranged, is anything but certain in its action, though, as we shall presently see, single non-flywheel pumps can be constructed with positive valve gear.

Duplex Pumps.

The duplex pump was invented by Henry R. Worthington, of New York. It was not until about the year 1880 that such pumps were generally introduced to this country by the company bearing the name of the inventor, but their good qualities gained speedy recognition. Duplex pumps are now constructed by most leading makers in all manufacturing countries.

The duplex pump comprises two single pumps arranged side by side on the one framing. The slide valve of each steam cylinder is actuated by a lever which is rocked by the movement of the piston rod of the adjacent cylinder. Each piston thus acts to give steam to the other, and on finishing its stroke it must wait for its own valve to be acted upon before it can renew its motion. This pause allows all the water valves to seat quietly and removes all harshness of motion. The pump is absolutely reliable in its action, for as one or the other of the steam valves must always be open, there can be no dead points. The machine starts directly the steam is turned on, and will continue to work until it is shut off.

It is particularly necessary for the proper working of duplex pumps that the piston and gland packings should be uniform. If such packings in one cylinder are tighter than in another, the one piston will move sluggishly as compared with its neighbour, with the result that the motion of the slow piston will be reversed before it has completed its stroke. One side of the pump will thus exhibit the effect known as "short stroking." As the clearance remains constant whatever may be the length of the pump stroke, it will be readily understood that short stroking means waste of steam.

Worthington Boiler Feed Pumps.

The sectional view through one side of a Worthington piston pattern feed pump, given at fig. 24 in our last chapter, clearly represents the arrangement of the separate admission and exhaust ports at each end of the steam cylinder, and the disposition of the suction and delivery valves above the water cylinder. In its motion each steam piston passes over its cylinder exhaust ports, and an efficient cushioning effect is thus obtained from the entrapped steam. Levers are provided (with the pump illustrated), to permit of the operation of the machine by hand. Pumps of the type shown are suitable for steam pressures up to 160 lb. per square inch. Similar pumps are constructed with compound steam cylinders.

Fig. 25 is an illustration of a Worthington ram pattern feed pump. A sectional view through one side of the pump is given at fig. 26. It has a great advantage over piston or plunger pumps, in that leakage can be at once observed and readily taken up whilst the pump is in motion by simply screwing up the glands. The water end of the pump is so designed that there are no air pockets when it is placed in a vertical position, and thus the machine can be fixed either horizontally or vertically as may be required. In the pressure pattern type Worthington ram pumps, for working against pressures up to 300 lb. per square inch, the water valves are located in separate valve boxes consisting of independent castings bolted to pump barrels. Such an arrangement permits of the ready inspection and renewal of the valves.

Mumford and Anthony's Patent Duplex Pump.

A sectional view through one side of the above-named pump (made by Mr. A. G. Mumford, Culver Street Engi-

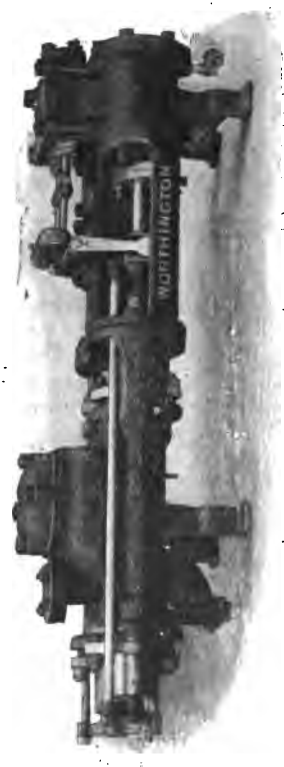


Fig. 25.

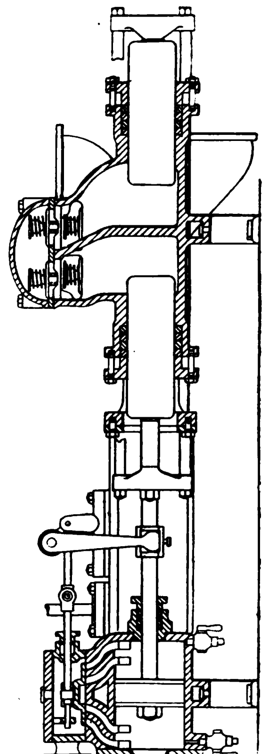


Fig. 26.

neering Works, Colchester) is given at fig. 27. The maker's general description is as follows :—

"In the steam cylinder there is only one moving part, viz., the piston, in which steam passages are so constructed

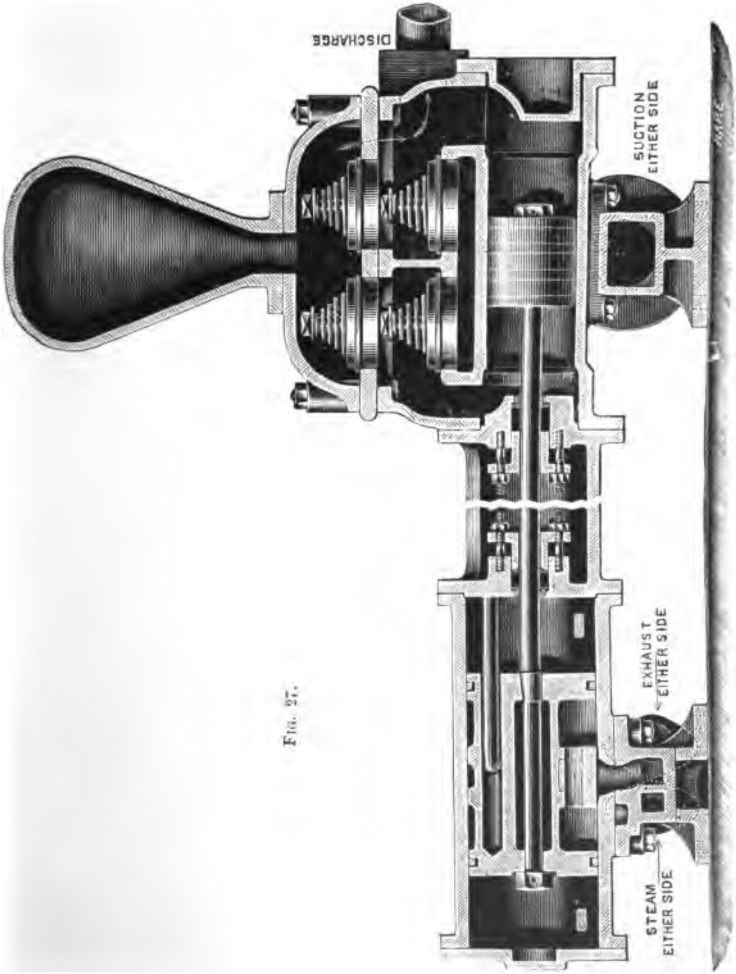

DISCHARGE
SUCTION
EITHER SIDE
EXHAUST
EITHER SIDE
STEAM
EITHER SIDE

Fig. 27.

that the usual slide valve is dispensed with, as well as levers, pins, and other small parts. The surfaces of the pistons are especially lengthened, so that their durability has become phenomenal. Each piston acts as a slide valve for the admission of steam to the opposite cylinder and (in due course of travel) its eduction passage, besides discharging its duty in the ordinary way of actuating the pump. The steam passages are carefully arranged, so that either piston will start from any position. There are no dead centres, and, throughout a course of severe tests and lengthened trials, these pumps have never been known to stick nor stop of their own accord. They are easy in working, and there is a complete freedom from shock. The pistons are perfectly balanced, and a certain steam cushion is provided at the end of each stroke. Full travel is always assured, and, in the event of pressure being withdrawn, the cushion is maintained."

Mumford's "Favourite" Donkey Pump.

We illustrated this well-known flywheel donkey pump (double acting) at fig. 22, in our last chapter. It is a very compact pump, and can be readily bolted to a boiler, to the side of a vessel, or other vertical support.

Mumford's Cameron-type Pump.

An illustration of the above is given at fig. 23 of our last chapter. The valves and all parts of this self-contained pump are adapted for working against heavy pressures, and are readily accessible for easy inspection.

The illustration shows a double-acting pump; but they are also made single acting—that is, with rams instead of with plungers or water pistons.

Weir's Single Feed Pumps.

The Weir patent direct-acting feed pumps, as they are termed, have been known for a number of years as high-class machines for marine work. Fig. 28 is an illustration of the Weir pump of the standard land-service type, whilst fig. 29 is a sectional view taken at right angles to fig. 28.

FIG. 28.

The makers (Messrs. G. and J. Weir Limited, of Cathcart, Glasgow) give the following specification of this machine:—

"The Weir standard feed pump is single cylinder, double acting, and vertical. The steam cylinder is of close-grained cast iron, and covered with planished sheet steel. The pump end is of cast iron, fitted with best Admiralty gun metal liner. The pump rod consists of one piece of cold-rolled manganese bronze. The valve seats and valve are of the Weir patent group type, which comprises a number of small valves in a circular gun metal seat, thus affording a large valve area with only a small lift. The valves are of special bronze, and the valve seats of Admiralty gun metal. The water pistons are also of gun metal, and fitted with ebonite packing rings. These pumps are suitable for boiler feeding up to 200 lb. pressure, and can pump either hot or cold water.

"The steam valve gear comprises a main and an auxiliary valve. The main valve is for distributing steam to the cylinders; the auxiliary for distributing steam to work the main valve. The main valve moves horizontally from side to side, being driven by steam admitted and exhausted from each end alternately. The auxiliary valve is actuated by lever gear from the rod of the pump, and moves on a face on the back of the main valve, and in a direction at right angles to the main valve. By this arrangement there is no dead centre, the action being absolutely positive, because the only possible position in which the main valve can rest is at full travel, either for an up or down stroke of the piston. Both the main and auxiliary valves are simply slide valves, but the former is half round, the round side working on the cylinder port face, which is bored out on one side to fit the valve. On the back of this main valve a flat face is formed for the auxiliary valve to work upon. Both ends of the main valve are lengthened so as to project beyond the port face, and are turned cylindrical with flat ends. Caps are fitted on each of these ends forming cylinders, which are closed at the mouths by the flat ends of the main valve, which act as pistons. The function of the auxiliary valve is to admit steam through the ports on the back of the main valve to move the main valve from side to side. The ports for

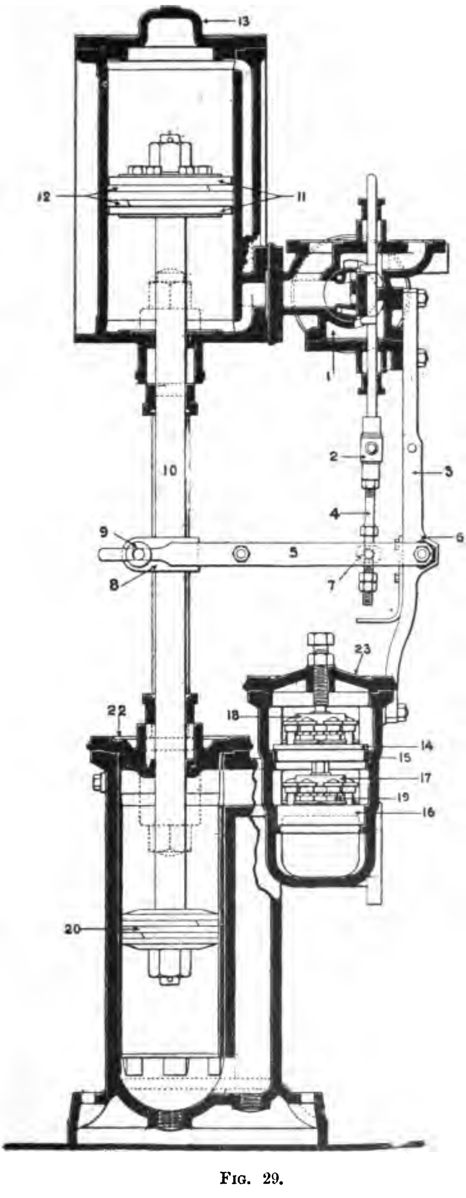


FIG. 29.

admitting steam to the top and bottom of the cylinder are arranged to cut off before the end of the stroke, and so slow down the pump, thus permitting the water valves to to settle quietly and relieve the connections from any shock. On the last quarter of the stroke the steam is thus used expansively, so effecting a considerable economy in steam consumption. Provision is made, however, by turning round the caps covering the end of the main valve for admitting live steam during the entire stroke, as, when the pumps are starting and the metal is cold, the steam condenses and it is necessary to clear out the chambers of water. These caps are turned by means of the gun-metal spindles with indicating pointers at each side of the steam-valve chest. When the pump is fairly started, these bye-passes—one for the up stroke and one for the down—are closed till the pump is working silently. The main and auxiliary valves are practically the only two moving parts in the valve chest. The stroke can be adjusted while the pump is working by the nuts on the valve spindle in accordance with the centre punch marks on the front stay, and is constant."

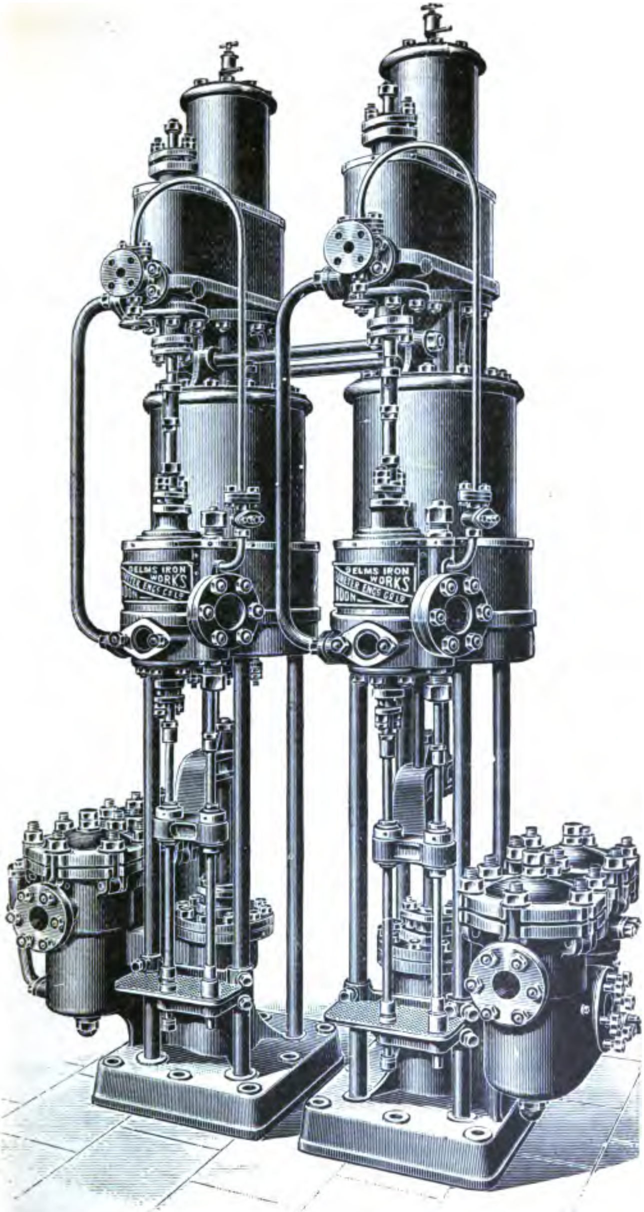
Messrs. Weir give the following particulars of two trials of one of their boiler feed pumps having a steam cylinder 8 in. diameter, pump 6 in. diameter, stroke 15 in., double acting:—

Steam pressure at pump	110 lb.	...	107 lb.
Water pressure at pump	164 lb.	...	164 lb.
Revolutions (or double stroke) per minute	15·9	...	6
Efficiency of pump....................................	97·25%	...	96·6%
Pounds of water delivered per pound of steam...	84·6	...	55·3
Pounds of steam per net W.H.P.....................	63·1	...	95·3
Foot-pounds of work per B.T.U. to 212 deg.	31·1	...	21

It should be noted that the "efficiency of pump" in the above particulars refers to the relation between the actual volume of water delivered and the theoretical displacement of the water piston. Thus, in the first trial the "slip of the pump was 2·75 per cent, and in the second trial 3·4 per cent.

The Pulsometer Co.'s Compound Feed Pump.

Fig. 30 is an illustration of a pair of "Karoome" compound feed pumps, constructed by the Pulsometer



4CP

FIG. 30.

Engineering Company Limited, of Nine Elms Ironworks, Reading. The makers state that these pumps can be worked with any pressure from 50 lb. to 350 lb. per square inch, that the steam used is less than 50 lb. per duty horse power per hour, and that, despite the fact of the water being in many cases delivered at a pressure 50 per cent above the steam pressure, the diameter of the high-pressure cylinder is considerably smaller than that of the pump barrel.

Bailey-Davidson Pumps.

Fig. 31 is a sectional elevation representing the "Davidson" double-acting pump, by Messrs. W. H. Bailey and Company, of the Albion Works, Salford.

The steam chest of the Davidson pump is bored out to receive the slide valve, which has a curved face, and also the double-headed trunk piece which assists in operating the

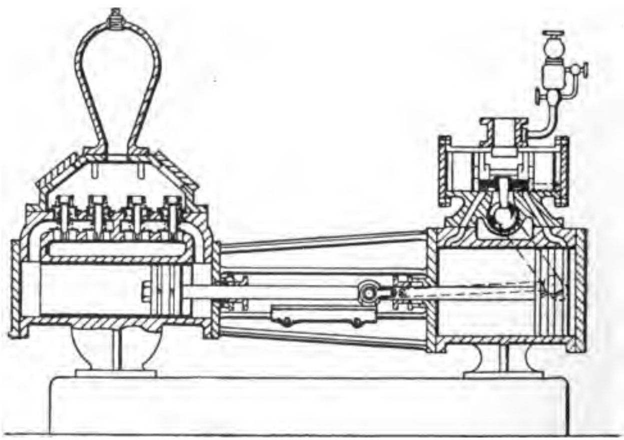


Fig 31.

valve. The valve is oscillated (for controlling the pressure on the trunk heads) by a cam arranged in the exhaust port, the rocking of the cam being effected by linkage worked from the piston rod as shown. The linear movement of the valve

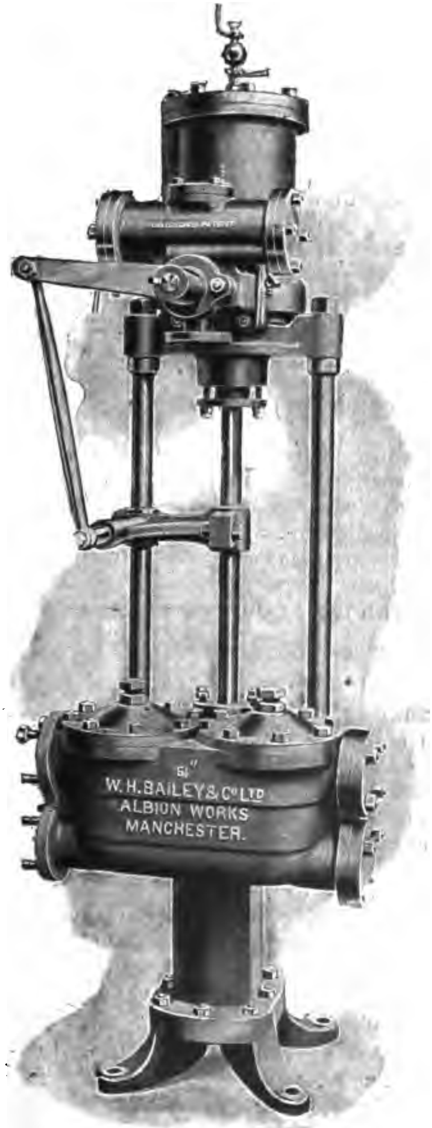


Fig. 32.

(for controlling the admission of steam to the cylinder) does not, however, depend entirely upon the steam admitted to the back of the trunk heads, for should that not be quick enough to operate the valve with the pump under a high rate of speed, the cam will itself carry the valve mechanically, and thus prevent the piston from striking the cylinder heads. This, the makers claim, is one of the most important features of the pump, the slide valve being as much under the control of the piston rod as is the valve of the ordinary steam engine worked by an eccentric, instead of being independently controlled by a trunk or like part operated only by the direct application of steam thereto, as is usual with many pumps of this class.

Fig. 32 is an illustration of Messrs. Bailey's vertical type of high-pressure boiler feed pump for marine or land service. As will be seen, it is of very compact and simple design, and the whole of the working parts can be readily got at for inspection and renewals.

Davidson steam pumps can be made with compound steam cylinders, and when required the makers also supply boiler feed pumps with triple cylinders.

Worthington Long Stroke Vertical Pump.

Fig. 33 illustrates one type or model (Type E) of the Worthington long stroke vertical feed pump. The following particulars are extracted from the makers' description :— The steam end is fitted with balanced piston valves. The valve motion can be adjusted externally, and whilst the pump is in operation, for varying the length of stroke. It is not necessary to alter the adjusting nuts except for very wide ranges of speed. But very slow running requires a later admission of steam in the cylinders than is the case with the pump running at high speed owing to the absence of momentum in the former case. The machine is designed for working pressures—both steam and water—up to 350 lb. per square inch. The water end of the pump is made in two separate parts, the valve box being separate from the water barrels and held firmly in place by studs. The suction and the delivery valve in each water valve chamber are both arranged about the one spindle. This arrangement is

FIG. 33.

followed throughout the pump to the extent necessary to the number of valves required to secure liberal valve area.

PEARN'S RAM PUMPS.

Fig. 34 illustrates a single ram Cameron type pump by Messrs. Frank Pearn and Company Limited, of West Gorton, Manchester. As is usual with this type of pump, the columns supporting the overhead steam cylinder are made to act also as air vessels. But the pump illustrated has its ram packed under what is known as Pearn's patent system of packing, which the makers describe as follows:—

"The illustration (fig. 34) shows the pump chamber in section. The ram A is of the ordinary type used in double-

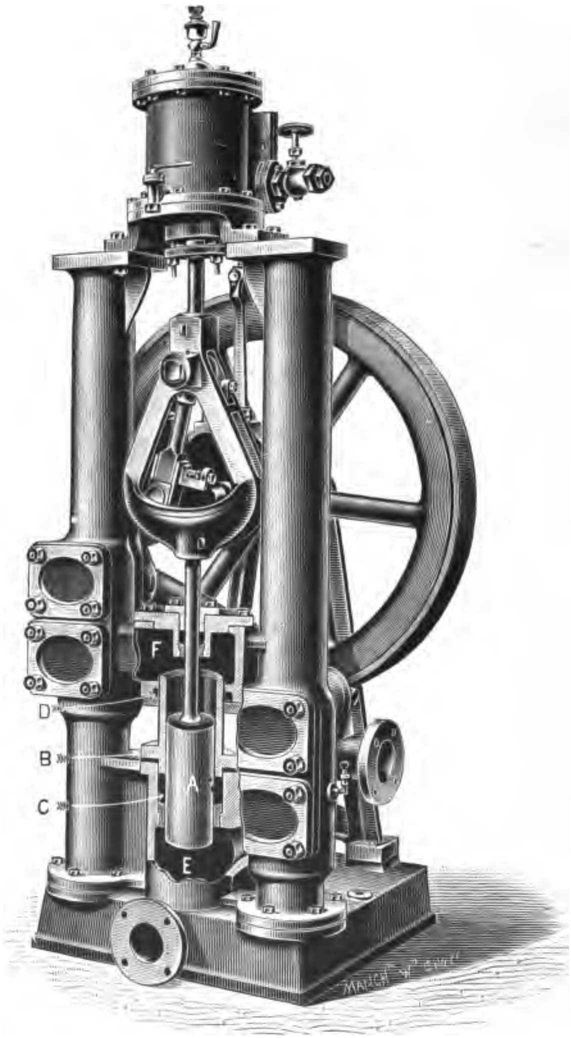


FIG. 34.

acting externally packed pumps, but made much shorter, as it works through one stuffing box only, instead of two, as in ordinary double-acting ram pumps.

"B, the pump gland, is the novel feature in this packing, as it not only performs the ordinary function of a gland, but is in addition the pump chamber F for one side of the pump. The collar for taking the gland bolts is screwed on after the gland is inserted through the top chamber F. C is the ordinary stuffing box, in which may be used spun yarn or any other packing. D is a joint ring to make a joint round the gland chamber B, and is slackened when the gland is required to be moved.

"The great advantages of this packing are, that a straight external packing is obtained in much less length than where two glands and stuffing boxes are used. All leakage is external, and shows at once when requiring re-packing. Leakage from pump chambers E to F is impossible. There is only one stuffing box; this reduces friction and is less expense and trouble in packing. The gland is guided at each end, and cannot be twisted by screwing one side harder down than the other, so any friction or wear due to such a cause is obviated."

Green's Ram Pumps.

Fig. 35 is a sectional view through the cylinder and steam chest of a ram pattern boiler feed pump fitted with Balkwill's patent slide valve, as made by Messrs. E. Green and Sons Limited, of 2, Exchange Street, Manchester, and of Wakefield. In the ordinary single-acting ram pumps, although water is delivered only on the down or in-stroke of the ram, live steam is supplied to each side of the piston.

With a pump fitted with this valve live steam is only used during the downward stroke in pumping the water, and the exhaust from the upper side of the piston is utilised to lift the "dead" ram during the upward or return stroke, and to overcome frictional resistance.

The valve works on a cylinder face having four ports, one to the top of the cylinder, one for exhaust, and two to the

bottom end. Live steam from the boiler is admitted to the top of the piston, cut off and expanded as in the case of a common slide valve, but at release, instead of the steam exhausting into the atmosphere or the condenser, both ends of the cylinder are put into communication for a short period,

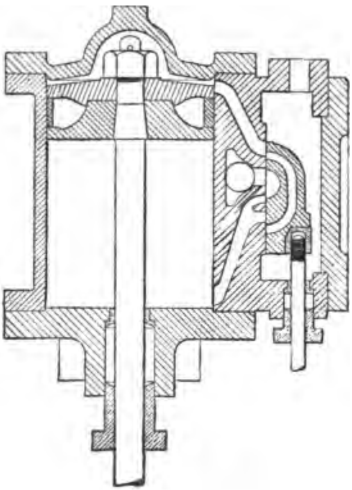


FIG. 25.

and then the ports at the bottom are closed, and the top opened to exhaust. In this way a portion of the steam used during the downward stroke passes to the bottom end of the cylinder, and on expanding performs work on the piston during the upstroke.

In addition to economy of steam, the patent valve has another great advantage over the one ordinarily used for the purpose. With the latter the speed of the piston and ram during the upstroke is considerably faster than at the downstroke, with the result that the pump valves are brought

too quickly down on their seats, thus causing unnecessary wear and tear; whereas with the patent valve the speed of both strokes is much more uniform, and the pump works with greater steadiness.

The following is a condensed summary supplied by the makers of tests made on two single-ram pumps, one fitted with a common slide valve, and the other with the Balkwill valve, but in all other respects similar in size and construction. Dimensions of pump: Diameter of ram, 6 in.; diameter of steam cylinder, 8½ in.; stroke, 8 in.

	Test No. 1. Ordinary valve.	Test No. 2. Balkwill valve.
Duration of trial..............minutes	30	42
Total number of revolutions	1625	2512
Revolutions per minute	54·2	59·6
Total weight in pounds of steam entering cylinder	258¼	189
Steam pressure on boiler side of starting valve, lbs. per sq. in.	92	101
Water pressure on delivery side ....pounds per square inch	123	119
Total quantity of water pumped..............gallons	1158	1970
Water delivered per pound of steam ....pounds	44·7	104·2
Steam per I.H.P. per hour ....pounds	84·5	49·9
Efficiency of pump..............	87%	96%

Economy in favour of new patent valve 42 per cent in steam consumption per I.H.P. During the tests the steam cylinders were not lagged, or in any way covered, to prevent condensation.

It should be observed that the "efficiency of pump" in the above table is the relation between the water delivered and the theoretical displacement of the ram. Thus in the first case the "slip" of the pump is 13 per cent, and in the second 4 per cent.

In comparing the above record of tests with that supplied by Messrs. Weir of tests on their own pumps, and which we have given in the description of the Weir feed pump, it will

be seen that in the later record the steam consumption is stated in terms of the water horse power, or the external or useful work done by the pump; whereas in the former it is stated in terms of the indicated horse power, or the work done in the steam cylinder. From the data given by Messrs. Green, we can, however, readily calculate the steam consumption in terms of the horse power on the water, or delivery, or output side of the pump.

Thus, on referring to the table, we find that the pump with the ordinary valve delivered 44·7 lb. of water against a pressure of 123 lb. per square inch. Now, taking the pressure at the base of a column of water 2·3 ft. high to be 1 lb. per square inch, it will be seen that the units of useful work accomplished by the expenditure of 1 lb. of steam is $44{\cdot}7 \times 123 \times 2{\cdot}3 = 12645{\cdot}63$ foot-pounds. Therefore, the steam consumption per pump or water horse power will be

$$\frac{33000 \times 60}{12645{\cdot}63} = 156 \text{ lb. per hour.}$$

In the same manner it will be found that the steam consumption per useful or water horse power in the pump, fitted with a Balkwill valve, works out as follows:

$$\frac{33000 \times 60}{104{\cdot}2 \times 119 \times 2{\cdot}3} = 69 \text{ lb. per hour.}$$

CHAPTER VII.

Mine Pumps.

As is well known, the first steam pumping engine was the first practical steam engine of any kind. For though the force of steam was referred to by ancient writers long before the Christian era, and though the Marquis of Worcester thanked Heaven for showing him "so great a secret of nature, beneficial to all mankind," it was reserved for Thomas Savery, a Cornish mining captain, to do the thing of which others had dreamed and written. He built his first "fire-engine," as it was termed, for mine drainage, rather more than two centuries ago, in the year 1698.

In the Savery engine—as with the modern form of such engines very largely and advantageously used under certain conditions at the present day—a vacuum is created in a closed pump chamber by the condensation of steam admitted within it. By the atmospheric pressure the water to be raised is then forced into the pump chamber, from which it is expelled on the next introduction of steam, and caused, by the direct pressure of such steam, to flow up and out of the discharge or delivery main.

In the Newcomen pumping engine, by which the Savery engine was superseded, the condensation of the steam necessary for the formation of a vacuum is effected in a cylinder and beneath a piston connected to a beam. The unbalanced atmospheric pressure on the top of the piston then effects the elevation of the opposite end of the beam with the pump rods and bucket for the formation of the vacuum within the pump chamber fixed down in the mine or pit, within suction distance of the water. The subsequent descent of the bucket and the duly weighted pump rods effects the discharge of the water through the rising main to the pit bank.

James Watt's improvements on the Newcomen pumping engine, comprising the formation of the vacuum in a separate vessel (the condenser), the introduction of the air pump to improve the vacuum, the use of steam instead of atmospheric pressure on the upper side of the piston, the steam jacket, and other well-known features, need no description.

The Savery pumping engine had of necessity to be placed bodily in the pit and within suction lift of the surface of the water to be raised. But with both the Newcomen and the Watt pumping engine there is a distinct and widely separated "steam end" and "water end," the former (or the engine proper) being arranged at the surface, whilst the latter (or the pump proper) must be fixed in the mine or pit.

In modern mine pumps we have, in the more general and ordinary practice, a steam and water end arranged to form the one complete machine, which, like the Savery engine, is placed entirely in the pit. But before proceeding to consider various types of modern mine pumps we will here refer to the

Duty of Mine Pumps.

For the purpose of comparing his engines with the best work obtainable from horses, James Watt fixed upon the unit mechanical horse power (33,000 ft.-lbs. per minute), which soon became and remains the recognised British standard. In like manner, to enable the mine owners to appreciate what he proposed to do for them, he expressed the capability of a pumping plant (including both the pumping engine and boiler) in terms of the number of millions of foot pounds of work that could be performed by the expenditure of 1 cwt. of coal.

The following figures may be taken as indications of the performances of ancient and modern pumping engines :—

Savery engines	5,000,000	ft.-lbs.	per cwt.	of coal.
Newcomen engines......	12,000,000	„	„	„
Watt engines............	80,000,000	„	„	„
Present day	150,000,000	„	„	„
„ „ (best)......	180,000,000	„	„	„

It will be seen that with the Savery engine the expenditure of coal per horse power per hour was—

$$\frac{33000 \times 60}{5000000} \times 112 = 44{\cdot}3 \text{ lbs.}$$

With the last result given on list the coal consumption per horse power per hour works out as follows:

$$\frac{33000 \times 60}{180000000} \times 112 = 1{\cdot}23 \text{ lbs.}$$

In comparing these results with those that may be obtained with other engines, running, it may be, at high speeds, and with the boilers and the whole plant specially arranged for the attainment of exceptional results, it must be borne in mind that the comparatively slow speed at which pumping engines are generally run is not conducive to great economy in steam consumption. We shall return again to the question of the results actually obtained from pumps, but we may here consider for a moment the total amount of heat energy available from the combustion of 1 cwt. of coal.

The standard calorific value or the total heat of combustion of 1lb. of best steam coal is generally taken at 14,700 British thermal units. The mechanical equivalent of one British thermal unit being taken at 772 foot-pounds, it follows that if the whole of the heat energy could be converted into useful work the quantity of coal required per horse power per hour would be—

$$\frac{33000 \times 60}{14700 \times 772} = 0{\cdot}175 \text{ lb.}$$

Thus the burning of but one pound of coal would yield nearly 6 horse power for one hour.

Types of Mine Pumps.

It has been well said that "it is very difficult to design and construct a steam pump that will satisfactorily meet the exacting requirements of mine pumping. The service

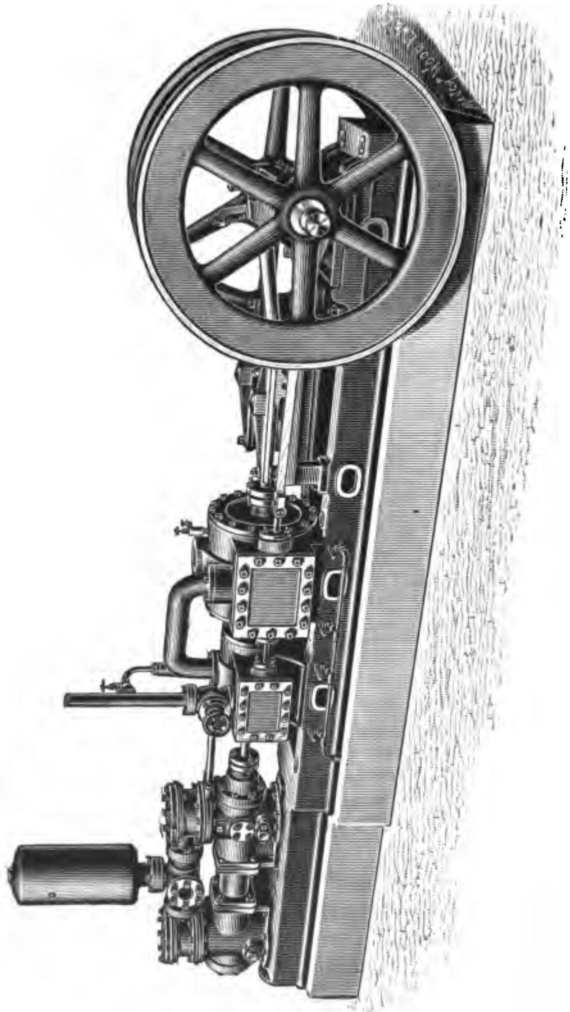


FIG. 36.

is generally rough, severe, and continuous. Great care must be exercised both in the selection and adaption of the material used in construction, as the water to be pumped is often of a kind that will attack and quickly destroy it. The location of the mine is usually remote from supplies, and any necessity for renewals or repairs, unless they can be made with unskilled labour and with little delay, may be attended with serious consequences."

PEARN'S COMPOUND RAM PUMPING ENGINE.

Fig. 36 illustrates the above-named crank and flywheel pump by Messrs. Frank Pearn and Co. Limited, of West Gorton, Manchester, suitable for use in collieries as an underground pumping engine. As will be seen, the external stuffing boxes of the rams can be readily packed, and all parts are easily accessible. The pump valves are of gun metal, and each is accessible by a separate door or cover.

The cylinders and pump end are fitted on the one base plate, ensuring the maintenance of accurate alignment of the working parts.

EVANS' "CORNISH" DOUBLE-ACTING RAM PUMPS.

Fig. 37 is an illustration of the above-named pumps as made by Messrs. Joseph Evans and Sons, of Culwell Works, Wolverhampton. The "Cornish" steam cylinder, as it is termed, which is adopted by Messrs. Evans for this and many other types of their pumps, is provided with a steam-operated controlling valve, but no tappets (or supplementary stalk valves which the main piston must operate on each stroke), are necessary, as with some other types of steam moved valves, and which are objectionable because of their liability to become stuck up or inoperative. Small ports are provided at each end of the cylinder which communicate with the back of a small plunger, working in a larger plunger arranged in the valve chest. As the main piston approaches the end of its stroke in either direction it uncovers one of the said ports to allow steam from the cylinder to act upon one end of the small plunger, and so move the same as to put the opposite end of the larger plunger in communication

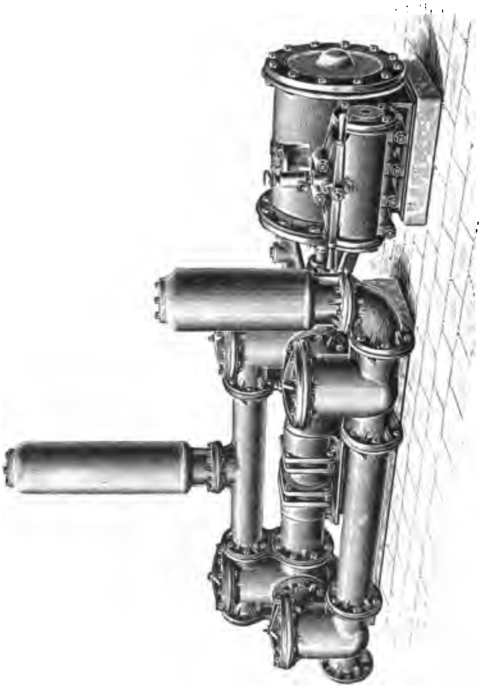


Fig. 37.

with the exhaust, and the adjacent end with the live steam in the valve chest. The large plunger, together with the slide valve, is thus carried over the main ports, thereby reversing the motion of the piston within the cylinder. The steam chest being placed on the side of the cylinder, and the bottom of the steam port on the same level as the bottom of

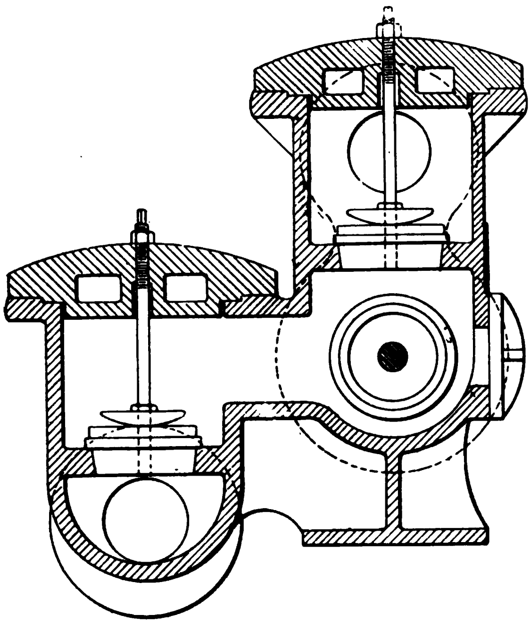


FIG. 39.

the cylinder, the makers state that the whole of the condensed steam is carried out at every stroke of the piston, whereby the necessity for drain cocks is avoided.

The pump end, as will be seen from the illustration, is provided with a ram working through a pair of stuffing boxes arranged in separate chambers. As there is a suction in one chamber and a delivery from the other, a double action is

obtained on every stroke of the single ram. The illustration shows both an air vessel on the delivery side and a vacuum chamber on the suction side.

Though Messrs. Evans can also make this type of pump, with a piston or bucket in place of the outside-packed ram,

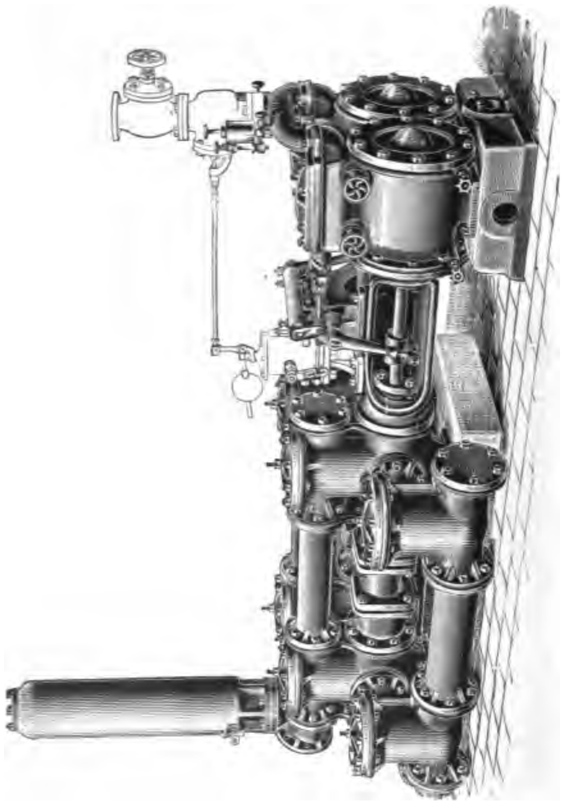


FIG. 38.

as illustrated, their recommendation is in favour of the latter, and they do not in any case advise the adoption of a piston or bucket pump for a waterhead of more than 300 ft,

on account of the loss of efficiency due to slip, which may become very great before detection when working with an internally-packed piston.

Evans' Duplex Pumping Engine.

This pump is illustrated at fig. 38. The water end comprises externally-packed rams with central stuffing boxes and cylindrical valve chambers, each valve having its own

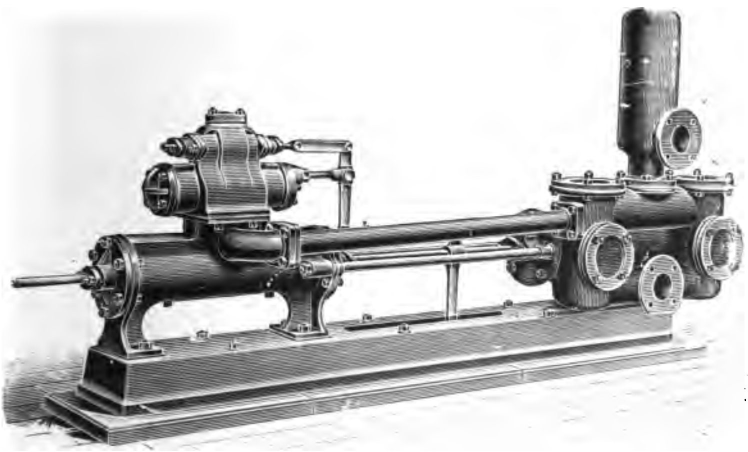


Fig. 40.

chamber. A sectional view through a pair of the valve boxes or chambers (one suction and one delivery) is given at fig. 39. The slide valves at the steam end are operated under the well-known duplex-lever system.

Evans' Hydraulic or Water Motor Pump.

This pump (illustrated by fig. 40) is constructed for working by water instead of steam. By utilising the motive power of a given head of water, a larger quantity of water may be thereby pumped to a relatively less height, or a relatively less quantity of water may be pumped to a greater

Fig. 41.

height than the head of motive column working the pump. It has been specially designed and constructed for draining "dip" workings in mines or collieries, and it will work quite satisfactorily when entirely submerged by water.

The makers state that they "supply these motors chiefly for underground use, in mines where water has to be pumped up from dip workings to the shaft bottom, and they are found to be very convenient for the purpose." They further state: "We have frequently applied these motors in positions where the water pipes have had to be carried 500 or 600 yards to the pumps, and we have them working in some cases under high pressures, up to nearly 2,000 feet head; on the other hand, we have also applied them in cases where the head of water to drive them has been as low as 25 ft., and forcing against a head of 300 ft. or 400 ft. During the colliery strike in the County of Durham, some years ago, a number of these pumps were left working down the pits for a period of three months; the water rose over them from various causes, chiefly because the main pumping engines were not kept regularly at work, but they still continued to work alone, and when the strike was over and the main engines set going these motors pumped themselves out."

Worthington Mine Pumps.

The Worthington compound mine pump, illustrated at fig. 41, is of the externally packed plunger or ram pattern. The two rams at the water end of the pump work through central stuffing boxes, as illustrated, into four separate and distinct water chambers, any one of which can thus be renewed. Pumps of this pattern are designed to safely withstand a working pressure of 300 lb. to the square inch, and the makers state that all the attachments are especially strengthened with a view to meeting the rough usage and hard work to which, in mine service, they are liable to be subjected.

Fig. 42 illustrates the Worthington triple-expansion mine pump (pressure pattern). This pattern is designed for use in cases where the head to be pumped against exceeds 300 lb. per square inch. The sub-division of the parts at the water

Fig. 42.

end is carried still further in this type, for, as will be seen, in addition to the construction of two part pump chambers with flange joint connections, the valves are arranged in a series of valve boxes or pot chambers. Each box contains

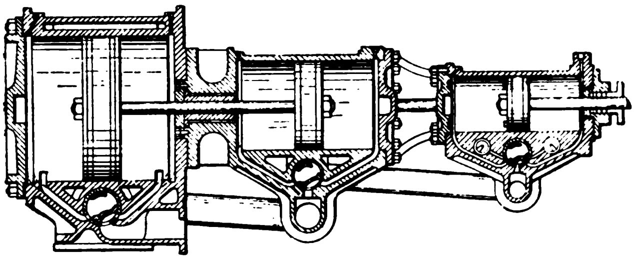


FIG. 43.

one or more small valves having a low lift, and which are easily accessible by screwing back the nuts on the eye bolts and removing the box covers.

Special features in this pumping engine are the patented method of connecting the three steam pistons, and the semi-rotative or Corliss-type valves. Fig. 43 is a vertical

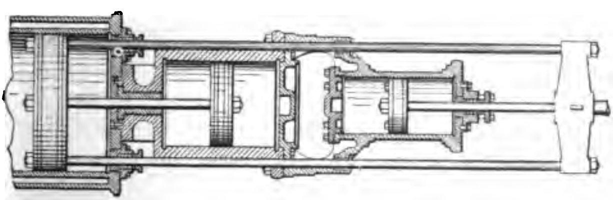


FIG. 44.

longitudinal section through one side of the steam end of the engine, and fig. 44 a horizontal longitudinal section of the same showing the arrangement of the piston rods. Each cylinder is provided with a single valve placed centrally beneath it, but the high-pressure cylinder has, in addition, a pair of adjustable cut-off valves. The valves serve to drain the cylinders, and by their own weight and

the pressure of steam on the unbalanced area, the valves are kept steam tight, and are free from the tendency to fall away from their seats which exists when inverted valves are employed, as is generally the case if they are placed beneath the steam cylinders. The two high-pressure cylinders (one on each side) are bolted directly to the cradles of the pump, with the intermediate cylinders next, but there is an intervening space between the adjacent covers of the high and intermediate pressure cylinders. The low-pressure cylinders are bolted directly to the intermediate cylinders, and are thus at one extreme end of the pump or pumping engine. The high-pressure piston rod is coupled directly to the pump rod. Between the high-pressure cylinder and the pump end there is a crosshead to which are attached two side rods connecting to the low-pressure piston, whilst the latter is connected by a central rod with the intermediate piston; the said central rod works through a long metallic sleeve made an exact fit to the rod, whereby the use of a stuffing box is avoided.

The makers claim the following advantages for their method of connecting the pistons:—

1. In this construction each of the three pistons and the interior of the cylinders are accessible by removing their respective cylinder heads, and each of these heads may be removed readily and without any interference with any piston rod or stuffing box.

2. That as each of the pistons is carried at the end of its rod, they may all be removed by simply slacking off their respective nuts.

3. In this construction the use of keys in the cylinder may be entirely avoided, and the effective area of the smaller pistons need not be reduced by rods larger than sufficient to carry their respective individual loads.

4. The direct course of steam between the high pressure and intermediate and low pressure cylinders in the order of expansion is retained, and in case of repairs, either the intermediate or the high pressure pistons and valves, or both, or the low pressure valves, can be removed and the engine run with the remaining cylinders or cylinder.

In the construction of their triple expansion pumps such

as illustrated, the usual practice of the Worthington Pump Company is to fit the low pressure cylinders with dash relief valves to regulate the length of stroke of the engine, and to steam jacket the intermediate and low pressure cylinders.

With regard to economy the makers give the steam consumption of their ordinary triple expansion condensing mine pumps at about 22 lbs. per useful or pump horse power per hour.

Arrangement of Condensers in Mine Pumps.

With pumps fixed in underground workings it is in many cases necessary to employ a condenser if only for the purpose of disposing of or "killing" the exhaust steam. For some services an air pump condenser either arranged independently or worked from the pump itself may be adopted. But it is frequently more convenient and sometimes essential to employ a surface condenser. In the water passing through the pump we have an ample supply of the condensing medium available. The surface condenser can be arranged either on the suction or the delivery side of the pump, so that the water flowing into or from the pump during its working is caused to pass through the condenser. The steam is thus effectually disposed of, and at the same time a vacuum is produced resulting in economical working of the plant. An air pump will be necessary and this may be either worked independently or be attached to and worked by the main pumping engine.

Bank or Surface Engines for Mine Pumping.

Both rotative and non-rotative pumping engines fixed at the ground level are employed for mine pumping. Of the former type the oldest and best known is the "Cornish." The name "Cornish," which is in these days adopted for quite small direct-acting non-rotative or non-flywheel pumps, was originally applied to the single-acting pumping engine invented and introduced by James Watt and his associates as an improvement upon the Newcomen. It comprises, as is well known, a single cylinder containing a piston connected

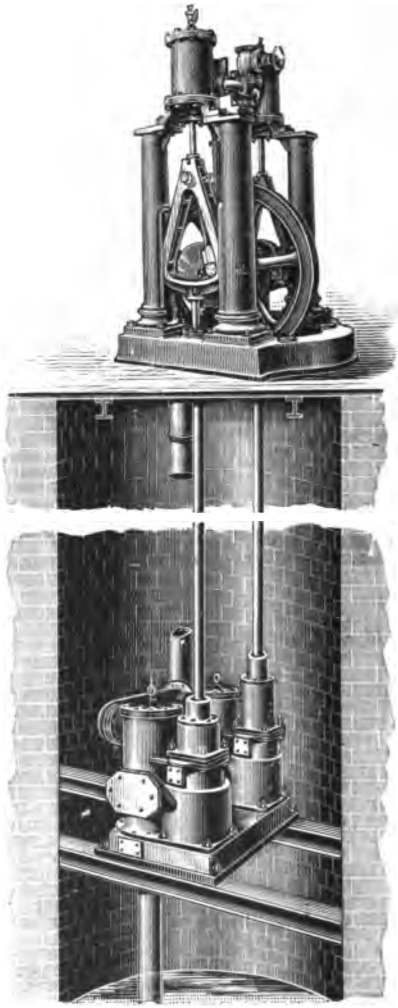


Fig. 45.

to one end of an overhead beam; to the opposite end of the beam the weighted pump rod is connected. The steam pressure acting upon the upper side of the piston raises the rod, whilst by the descent of the latter the water is forced through the rising main and the piston drawn to the top of the cylinder in readiness for the next cycle of operations.

Such engines are still constructed, and by compounding Mr. Henry Davey (of Messrs. Hathorn Davey and Company of Leeds) states that he has successfully employed a steam pressure of 150 lb. per square inch for working a Cornish engine.

Fig. 45 illustrates a small mine pump or well pump, as made by Messrs. Frank Pearn and Co. Limited, of West Gorton, Manchester, having the steam motor or engine at the ground level directly connected with the pumps fixed in the well or pit. The engine is of the "Cameron" type, whilst the pumps, which have externally packed rams, can be so arranged that their rods are in tension (instead of in compression, as is usual) during the delivery of the water.

An old non-rotative type of mine pump is the "Bull" engine, in which there is no overhead beam, the cylinder at the bank or surface level being placed directly over the pump proper in the mine; the piston rod can thus be directly connected to the pump rod. The system of working, or the cycle of operations, is the "Cornish."

Modern surface engines of the non-rotative type for mine pumping are frequently arranged horizontally, the motion of the piston being transmitted through a rod to one end of a rocker, angle bob, or quadrant pivoted adjacent to the pit, and having the rod from the pump fixed below connected to its opposite end. In the Davey pumping engine a pair of quadrants or angle bobs working separate pumps are employed, and these are so coupled together that the one serves to balance the other.

SINKING PUMPS.

Sinking pumps for suspending in a shaft during construction, for recovering flooded mines, and for similar services are made by most large pump makers. Such machines

FIG. 46.

must be very compact in design so that the space occupied shall be reduced to a minimum, and capable of withstanding the very rough handling they are sure to experience in service. Fig. 46 illustrates a differential ram (outside packed) sinking pump by Messrs. Joseph Evans and Sons, of Wolverhampton. A sectional view of the water end ot

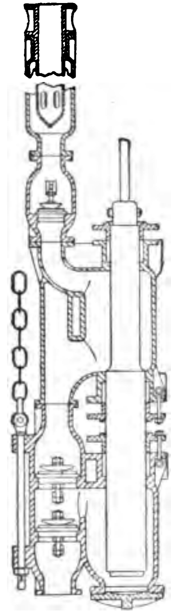


FIG. 47.

such a pump is given at fig. 47. The differential ram acts in the same way as the well-known bucket and plunger, or ram and bucket pump, in that although the pump is single acting on the suction side, it is double acting on the delivery side, for there is a discharge on each stroke. On the up or suction stroke of the ram the amount of water flowing into the lower pump chamber through the suction valve equals

the area of displacement of the larger end of the ram. On the down stroke this water is displaced from the lower chamber and forced through the delivery valve into the upper chamber, where the displacement equals only the difference between the areas of the two ends of the ram, such difference being usually equal to one-half the area of the larger end, or one end of the ram has twice the cross sectional area of the opposite end. Thus one-half of the water is discharged from the upper chamber on the down stroke, and the other half on the up stroke of the ram, and the work is thereby equalised as in a double-acting pump.

The pump illustrated is suitable for heads up to 600 ft., and the steam cylinder for steam pressures up to 100 lb. per square inch. The valve motion at the steam end is the Evans' and Tonkins' Patent, and is such as previously described in connection with another pump (fig. 37) by the same makers.

CHAPTER VIII.

Hydraulic-pressure Pumps.

Hydraulic machinery may be worked with a pressure of but a few pounds, or as many tons per square inch. But the term "pressure pattern" or "hydraulic pressure" is not generally applied to pumps working at pressures under 200 lb. per square inch. Pumps working at pressures of from 1 to 3 tons are generally described under the class or heading of "high-pressure hydraulic machinery."

A greater intensity of pressure than 3 tons per square inch is seldom adopted in practice, owing chiefly to the cutting or erosive action of the water, due to the enormous velocity at which it will be propelled by such a pressure through the valve apertures and connections. When it is remembered that a pressure of 3 tons per square inch will produce a velocity of about 1,000 ft. per second, or 11 miles per minute, there will be no difficulty in appreciating the importance of keeping the water quite clean in high-pressure

hydraulic work. The presence of grit will set up a most destructive sand-blast action as the water rushes through the valve ports and the connections generally. And it must also be remembered that where cylinders are employed, having a great thickness of metal to resist the high pressure within, the stress is not equally distributed, for the metal nearer to the interior will bear a greater proportion of the load than that at the exterior. Cylinder castings, because

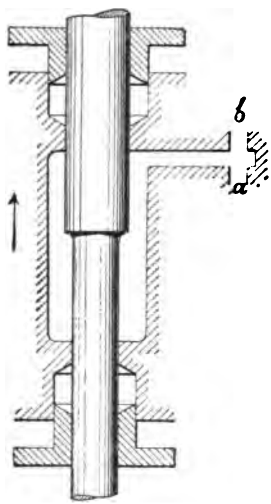


FIG. 48.

of such unequal distribution of the stress set up by the internal pressure, have been known to fail by the gradual cracking of the metal from the interior towards the exterior.

When high pressures are employed only a very moderate quantity of water is usually required, and the cylinders and rams or plungers at the pump or water end of a steam pumping engine for such hydraulic services are, therefore, of but small dimensions. It may be, indeed, that the rams

are so small as to render it somewhat difficult to provide sufficient rigidity, and to effectually pack the glands through which they work. A differential ram, such as illustrated in the sectional sketch diagram at fig. 48, may then be advantageously adopted. As will be seen, the ram has two diameters, so that on its up stroke, or its movement in the direction indicated by the arrow, a vacuous space is formed in the cylinder, and the water thus enters through the suction connection *a*; whilst on the down stroke there is a corresponding displacement, whereby water is forced out from the cylinder through the delivery connection *b*. Both the suction and delivery connections are, of course, provided with suitable valves. With only a small difference in the

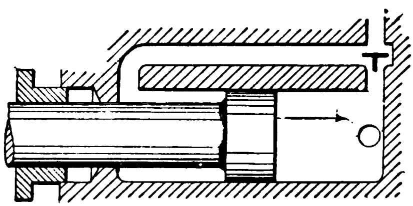


FIG. 49.

areas of the respective ends of the ram, there will be but a small discharge on each pump stroke, and thus we can provide ample cross-sectional area throughout the ram to resist any desired intensity of pressure, without affecting the quantity of water delivered.

Fig. 49 is a sketch diagram of the ram and piston type of water end frequently employed in rotative hydraulic pumping engines, for the purpose of obtaining the advantages of the equalisation of the work, and a steady flow resulting from a double action on the delivery side, with the simplicity of a single action on the suction side. The same result is thus obtained as with the differential ram sinking pump, illustrated at fig. 47 in our last article. The piston must, of course, be packed with hemp or the like, or with self-acting leathers. Our fourth chapter contains some particulars concerning packing.

Pearn's Hydraulic Pumping Engines.

Figs. 50, 51, and 52 illustrate three types of hydraulic pumping engines, as constructed by Messrs. Frank Pearn and Co. Limited, of West Gorton, Manchester.

In the type illustrated at fig. 50 there are three single-acting rams, each of which is directly connected by a yoke piece to the piston rod of the steam cylinder mounted above it. The crank shaft, which operates the steam slide valves, is driven from the yoke after the manner of the "Cameron" type boiler feed and general service pump. The makers give the following description:—

"This type of pump is largely used for working accumulators, &c. Having three rams, the delivery is very continuous when working direct without an accumulator. It is very suitable for presses, cranes, hoists, lead presses, &c., &c. It is made in several sizes to work at pressures from 700 lb. upwards. It is very compact, and all parts are easily accessible. The crank shaft is made of forged steel, and works in gun-metal bearings, all adjustable; the connecting rods are of best hammered iron, and are fitted with gun-metal bearings at each end, all adjustable."

The horizontal type rotative or flywheel hydraulic pumping engine, illustrated at fig. 51, affords a practical example of the ram and piston type. The makers give the following description:—

"This pumping engine is designed for working hydraulic presses, hoists, cranes, winches, capstans, &c., either direct or in connection with an accumulator. It is made in numerous sizes, either high pressure, compound, or condensing, and to work against pressures up to 1,000 lb. per square inch. The pumps are single acting on the suction, and double acting on the delivery. They are brass lined, and fitted with gun-metal pistons with cup leathers."

Fig. 52 is an illustration of a hydraulic ram type pumping engine, suitable for very high-pressure work. Each pump or water end is provided with a pair of rams which are single acting, but being coupled together by outside rods, and yoked to the one piston rod, a double action is obtained, for there is both a suction and a delivery action on each

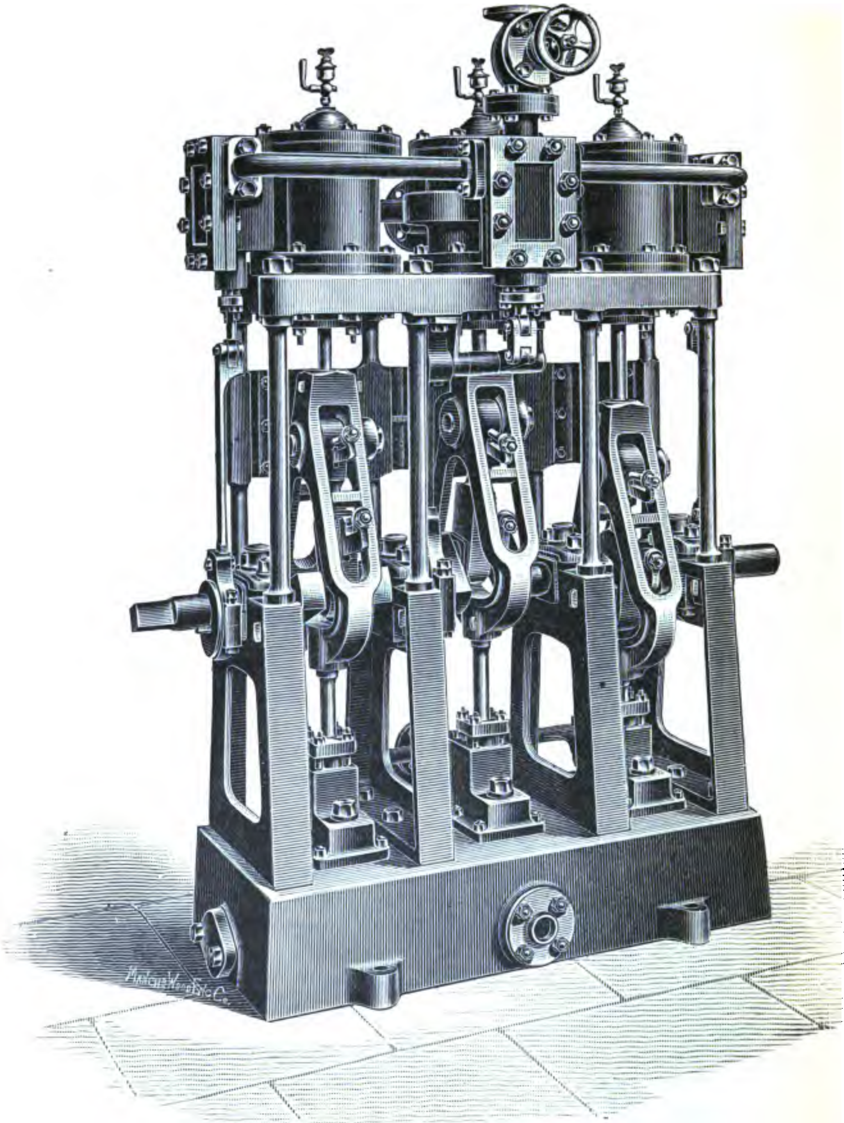


FIG. 50.

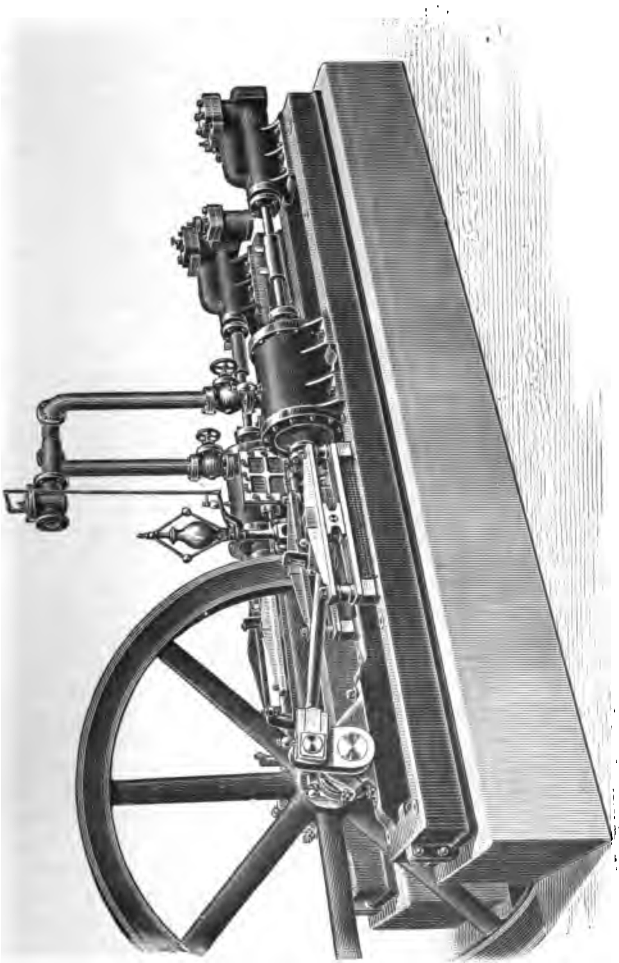


Fig. 51.

stroke of the steam piston. The complete engine is made up of two steam cylinders, each connected with a water end carrying two rams. Thus on each revolution of the crank shaft we have four suction actions and as many deliveries, or in other words, we have a four-throw pump action ensuring a continuous flow through the discharge pipe. The makers give the following description:—

"This is a very high-class type of engine, specially designed for high speeds and heavy pressures up to three tons per square inch. The pumps are made of solid forged steel; all the passages, valve chambers, and pump barrels are bored out. The rams are made of phosphor bronze; they are outside packed, of easy adjustment, and supported at each end by suitable crossheads. The steam cylinders are cleated with sheet steel, and fitted with sight-feed lubricators, steam stop valve, and drain cocks. The engine can be worked either direct or through an accumulator."

Rice and Co.'s Hydraulic Pumping Engine.

Fig. 53 is an illustration of a rotative hydraulic pumping engine, as constructed by Messrs. Rice and Co. (Leeds) Limited, of Neville Works, Elland Road, Leeds. The makers give the following description:—

"The illustration shows our standard type of steam-driven pumping engine, driving direct on to four single-acting pump rams. These engines are strongly constructed to suit the heavy and continuous work they have to do. They are provided with ample bearing surfaces throughout, so that they run very smoothly, and the wear is inappreciable. The bed plate is an iron casting of strong section, having the crank-shaft bearings cast solid with it, and for convenience of transport the larger sizes are made in halves, which are tongued and bolted together, making a very rigid connection. The crank arms are of mild cast steel, shrunk and keyed on to a forged steel shaft running in phosphor-bronze bearings. The crank-shaft bearings are made adjustable in the horizontal direction. The connecting rods, valve spindles, etc., are of mild steel. The cylinders are of hard close-cast iron, and the pistons are packed with cast-

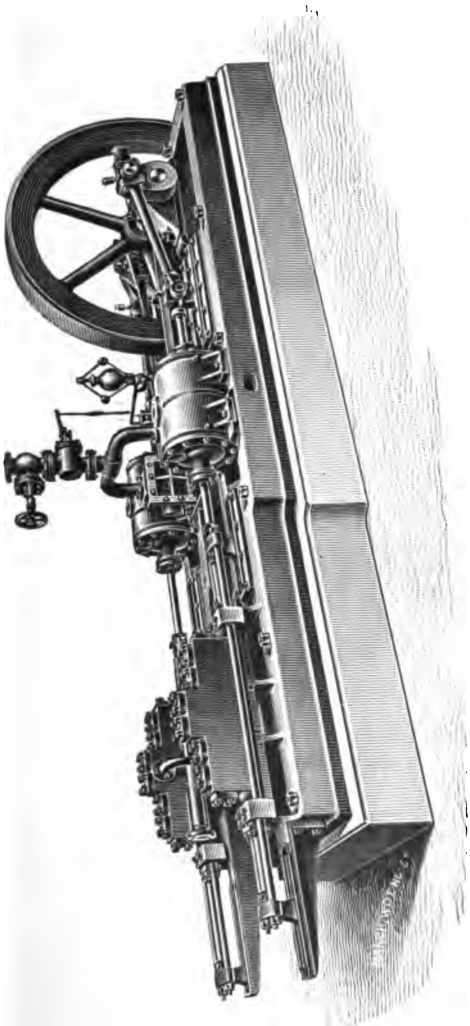


FIG. 52.

iron rings and Oldham's patent steel spiral coils. The pump bodies are of a very strong mixture of cast iron, and are fitted with phosphor-bronze glands and bushes. The valve and valve cases are of phosphor bronze, and made with good free openings for the suction and delivery. These engines are controlled by a Porter loaded governor, which is coupled to the equilibrium valve lever. This lever is arranged so that, when pumping into an accumulator, the accumulator can be connected to it in such a way that the engine is stopped as soon as the loaded ram reaches the top of its stroke, and started again as soon as the ram begins to descend, thus keeping the supply of water completely under control and economising steam. The cylinders are covered with cold-rolled steel sheets, and fitted with drain cocks and lubricators, and the whole engine is well finished throughout. The pumps are supplied complete as shown, and are made suitable for a working pressure of 1,500 lb. per square inch (100 atmospheres), but can also be made for any working pressure required."

Worthington Hydraulic Pump.

The non-rotative hydraulic-pressure pump, or pumping engine, as constructed by the Worthington Pump Company, is built on similar lines to the triple-expansion mine pumping engine by the same makers, described in our last chapter, and illustrated by the fig. 42 therein. Each pump or water end is provided with a pair of externally-connected rams, and with separate chambers for the valves. The engines are supplied for services up to 8,000 lb. (about 3½ tons) per square inch, and with simple, compound, or triple-expansion steam cylinders.

Worthington hydraulic-pressure pumps are for certain purposes sometimes bolted directly to a steam accumulator. The accumulator consists of an ordinary steam cylinder (such as would be used in a steam engine, but without the usual ports or valves), combined with a ram cylinder similar to that of a weighted accumulator. But no weights are employed, as the steam pressure acting on the piston within the first-named cylinder imposes the desired force on the ram, the latter being securely bolted to the piston.

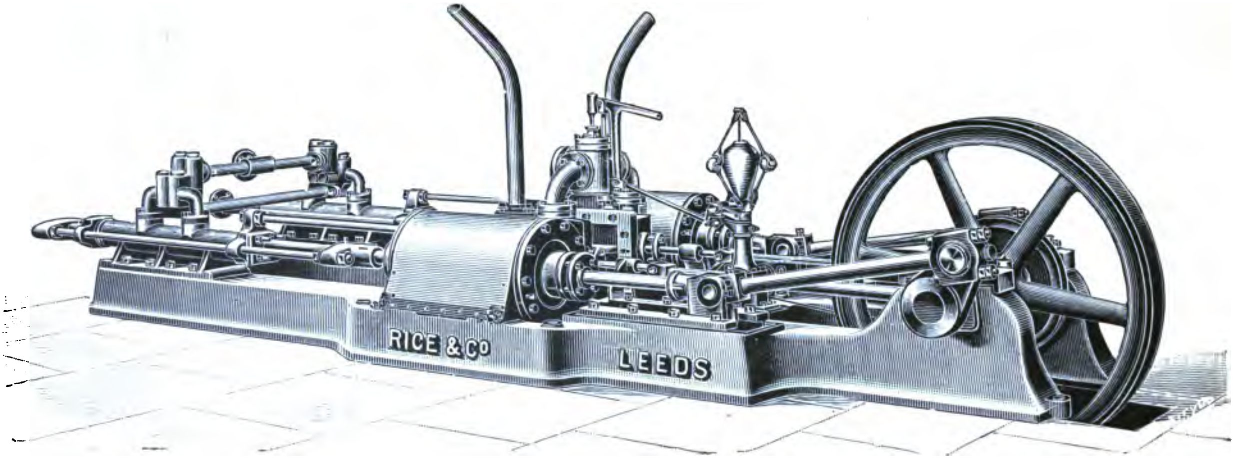


FIG. 53.

The makers state that "the advantage of this accumulator over the ordinary weighted accumulator can be readily understood. The most important and marked difference is in the effect on the water pressure. The steam accumulator is not subject to the tremendous shocks and jars due to the momentum of the weights of the weighted accumulator. The moving parts of the steam accumulator are so light that momentum is not a factor. The machine is very sensitive, and the variations of the water pressure very slight. The steam accumulator can be adapted to any space, as it can be placed either horizontally or vertically, and as it is not heavy it can even be hung from roof beams if necessary."

The makers further state that Worthington combined steam pumps and accumulators, as aforesaid, have been extensively used on the cruisers and battleships of the United States for general hydraulic service and for working the gun carriages and turrets.

CHAPTER IX.

Modern Savery-type Pumps.

Reference was made, in our article dealing with mine pumps, to the original Savery pumping engine of the year 1698, in which the steam is employed for the creation of a vacuum in the pump chamber, and by its contact with and direct pressure on the water, subsequently entering the chamber, for forcing such water through the delivery or rising main.

In the original Savery engine the condensation of the steam within the chamber was effected by the application of cold water to the exterior of the vessel, but in the modern form the water passing through the pump is the condensing medium. There is no exhaust steam to be dealt with, either in the original or the modern Savery engine.

The Savery-type engine, as now made, is known by various fanciful names, and is sometimes described as pistonless, as

pulsating, and as a steam vacuum pump. But as pumps of a totally different type may have the same descriptive terms applied to them, the appropriation of such terms by makers to a particular pump of their own is productive of considerable confusion. Just as "Otto-type" is the recognised name for gas engines working on the cycle originated by Dr. Otto, so should the type of pump in which the steam for performing the work is admitted into the pump chamber itself, to act directly upon the water, be associated with the name of the Cornish mining captain by whom it was invented and first applied more than two centuries ago.

We give hereunder some particulars of the various forms of the Savery-type pumps, under the names and with the aid of the descriptions and illustrations supplied by the makers themselves :—

THE PULSOMETER.

A sectional view of this well-known pump, made by the Pulsometer Engineering Co. Limited, of Nine Elms Ironworks, Reading, is given at fig. 54. The makers describe it as consisting "of a single casting called the body, which is composed of two chambers A A joined side by side, with tapering necks bent towards each other, and surmounted by another casting called the neck J accurately fitted and bolted to it, in which the two passages terminate in a common steam chamber, wherein the ball valve I is fitted so as to be capable of oscillation between seats formed in the junction. Downwards, the chambers A A are connected with the suction passage C, wherein the inlet or suction valves E E are arranged. A discharge chamber, common to both working chambers, and leading to the discharge pipe, is also provided, and this contains one or two valves F F, according to the purpose to be fulfilled by the pump. The air-chamber B communicates with the suction. The suction and discharge chambers are closed by covers H H accurately fitted to the outlets by planed joints, and readily removed when access to the valves is required; in the larger sizes handholes are provided in these covers. G G are guards which control the amount of opening of the valves E E. Small air-cocks are screwed into the cylinders and air-chamber, for

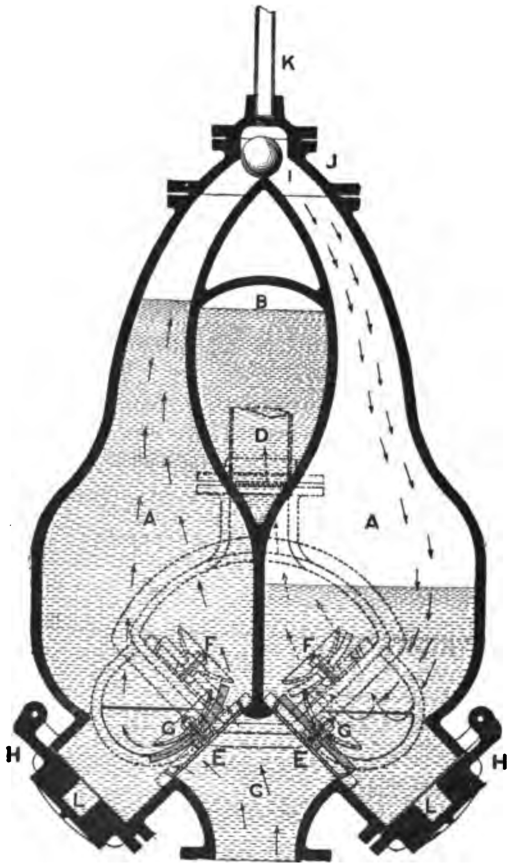


FIG. 54.

use as will be hereafter described. These are the general outlines of the construction of the apparatus, and they are sufficient for the understanding of the nature of its operations.

"The pump being filled with water, either by pouring the same through the plug-hole in the chamber, or by drawing the charge, as can readily be done by attention to the printed directions, is ready for work. Steam being admitted through the steam pipe K (by opening to a small extent the stop-valve), passes down that side of the steam neck which is left open to it by the position of the steam ball, and presses upon the small surface of water in the chamber which is exposed to it, depressing it without any agitation and, consequently, with but very slight condensation, and driving it through the discharge opening and valve into the rising main.

"The moment that the level of the water is as low as the horizontal orifice which leads to the discharge, the steam blows through with a certain amount of violence, and being brought into intimate contact with the water in the pipes leading to the discharge chamber, an instantaneous condensation takes place, and a vacuum is in consequence so rapidly formed in the just-emptied chamber that the steam ball is pulled over into the seat opposite to that which it had occupied during the emptying of the chamber, closing its upper orifice and preventing the further admission of steam, so allowing the vacuum to be completed; water rushes in immediately through the suction pipe, lifting the inlet valve E, and rapidly fills the chamber A again. Matters are now in exactly the same state in the second chamber as they were in the first chamber when our description commenced, and the same results ensue." The makers further state that the "change is so rapid that, even without an air vessel on the delivery, but little pause is visible in the flow of water, and the stream is, under favourable circumstances, very nearly continuous. The air-cocks are introduced to prevent the too rapid filling of the chambers on low lifts and for other purposes, and a very little practice will enable any unskilled workman or boy so to set them by the small nut that the best effect may be produced. The action of the

FIG. 55.

steam ball is certain, and no matter how long the pump may have been standing, it will start as soon as dry steam is admitted."

In the illustration at fig. 54 the pump is shown fitted with grid valves having rubber discs. For services where the action of the water to be pumped is detrimental to rubber, the makers employ clack valves consisting of a hinged iron plate or body with hickory seats, though it is to be noted that such valves are somewhat noisy.

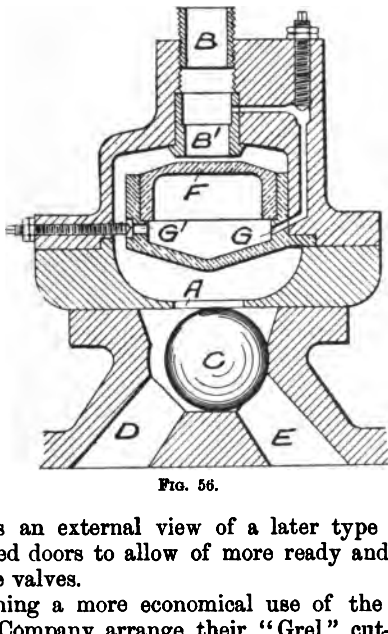


FIG. 56.

Fig. 55 is an external view of a later type pulsometer, having hinged doors to allow of more ready and convenient access to the valves.

For attaining a more economical use of the steam the Pulsometer Company arrange their "Grel" cut-off valve at the upper end of the pump. A sectional illustration of the upper end of a pulsometer fitted with a Grel valve is given at fig. 56. The makers thus describe the arrangement:—

"The operation of the combined valve is as follows: Commencing at the moment when the left-hand chamber D

is full of water, and the distributing valve C has moved to close the right-hand chamber E and open the left-hand chamber D, the expansion valve being open, the full steam pressure enters the left-hand chamber D and partially empties it. During this time, steam has been entering the special chamber F through the orifices G and G^1, thereby increasing the pressure therein, and as the water in the body of the pump falls there is a reduction of pressure without this chamber, the effect of which is to cause the movable part F of the expansion valve to rise and cut off the steam by closing the steam pipe. The expansion of the steam and the expulsion of the water continues until the chamber is nearly emptied, when the difference of pressures in the pump chambers brings over the distributing valve C. By this time the escape of the steam from the chamber F permits the pressure of steam in the steam pipe B to depress the movable part of the valve, and the steam rushes in to expel in turn the water which has flowed into the right-hand chamber during the emptying of the left-hand chamber, the correct working of the valve depending on the proper manipulation of the regulating screws."

THE "AQUA-THRUSTER."

Fig. 57 represents an external view, and fig. 58 a sectional view, of this pump, which is made by Messrs. W. H. Bailey and Co. Limited, of Albion Works, Salford. As will be observed, the spherical or ball valve for controlling the admission of steam in the pump last considered is in this machine replaced by a rocking plate or disc. In connection with the sectional view, the makers supply the following description :—

"The left-hand chamber is filling whilst the right is being emptied by the steam which is being admitted by the valve in the head of the pump. By the time the right-hand chamber is emptied the left one has filled ; the valve is drawn to the right, and the steam now forces the water out of the left-hand chamber and through the delivery valves, and water rushes into the right-hand chamber through the suction valves."

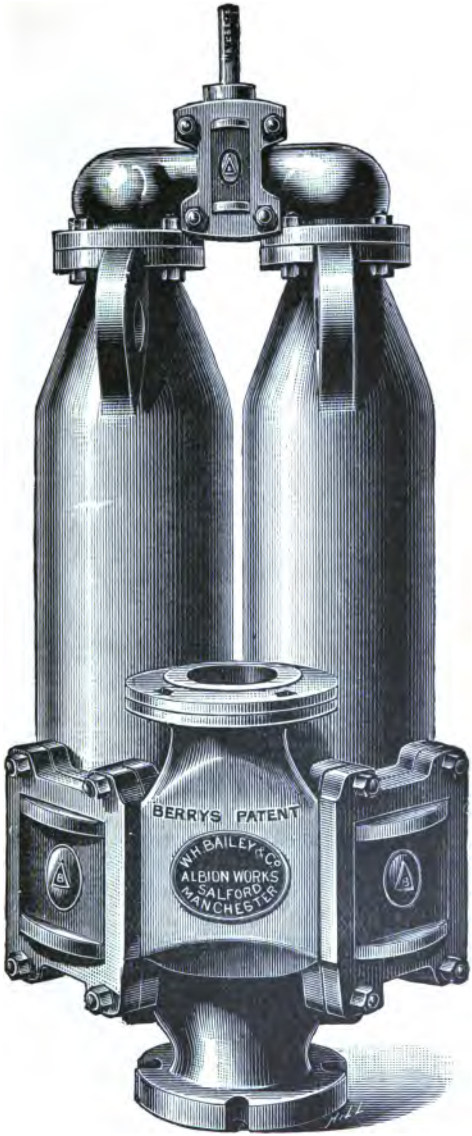

BERRYS PATENT
W.H. BAILEY & Co
ALBION WORKS
SALFORD
MANCHESTER

Fig. 57.

Fig. 58.

In their trade lists, the makers give the capacity of their pumps in the number of gallons of water lifted a total height of 32 ft. with 60 lb. steam pressure per square inch, stating, however, that a lesser quantity can be raised to such a height with 25 lb. to 30 lb. steam pressure.

THE "FLUOMETER."

This pump, by Messrs. Joseph Evans and Sons, of Wolverhampton, is illustrated in its original form at fig. 59. Fig. 60 is a sectional view of a later pattern.

In this pump the steam controlling valve is dispensed with. A is the working chamber, B the suction, and C the delivery valve. The action is described as follows:—

"The steam flowing in through the steam pipe depresses the water in the working chamber A, driving it through the delivery valve C, and up the delivery pipe; this is done without disturbance of the level, consequently the top layer of water in immediate contact with the steam becomes heated, and by reason of its specific gravity remains at the top, thus preventing further condensation, a portion of the water passing into the injection chamber E through the inlet K, where it remains under a pressure equal to that in the working chambers. In this manner the water level falls until it arrives at the off-set J, when a violent disturbance takes place, and a reduction of pressure is in consequence brought about. The injection water now rushes in from the chamber E, and completes the vacuum, causing a fresh supply of water to enter from the suction pipe. The inrush of the suction water is so violent that it is necessary to restrict the passage way into the pump at the lower extremity of the working chamber and to take in a small amount of air through the snift valve G. This air serves the double purpose of cushioning the flow of the water, and, in subsequently mingling with the steam, prevents condensation of the latter by reason of its low conductivity. The working chamber A being now filled again, the action is repeated as long as the steam remains on and there is water to pump."

STEAM CONSUMPTION IN SAVERY-TYPE PUMPS.

The great advantage of these pumps is their extreme simplicity of construction and their general handiness. They will work as well hung from a chain as if permanently fixed, and pump mud, slurry, and water containing a considerable quantity of solid matter, such as would quickly

FIG. 59.

render useless the wearing parts of other pumps. As a set-off against this, we have what is sometimes referred to as the "steam-eating" power of these pumps. But though

they undoubtedly have a considerable appetite, the steam consumption of Savery-type pumps is not so enormous as is sometimes assumed, and indeed, under certain conditions, they will compare favourably in this respect with ordinary

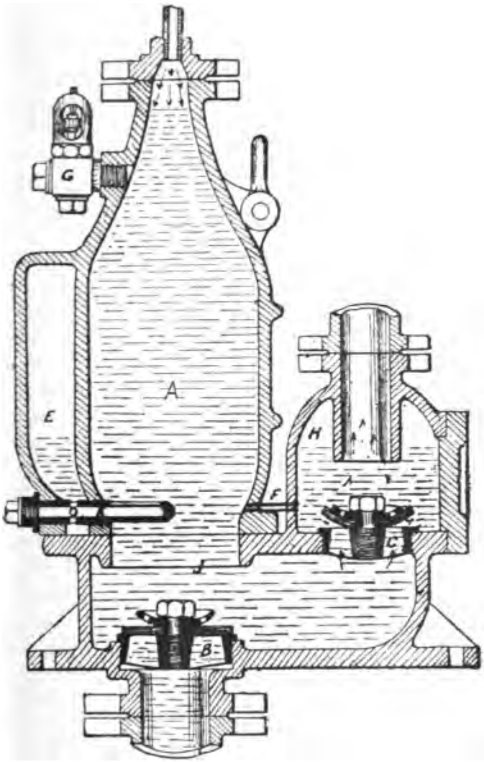


FIG. 60.

pumps in which pistons and rams or plungers are interposed between the steam and the water which it raises.

Reference was made in Chapter V. to a paper read before the Institution of Mechanical Engineers in 1893, by

Mr. A. Borodin, on the working of steam pumps in Russia. The author of that paper gives particulars of a test of a small Savery-type pump (of less than one-half actual horse-power) showing a steam consumption of 860 lb. per pump or useful horse power per hour, or 2,300 foot-pounds per pound of steam. In the discussion on the paper, one member gave the steam consumption, obtained by a rough calculation from the performance of a Savery-type pump he had employed, as 4,300 foot-pounds per pound of steam, which is equal to 460 lb. of steam per pump horse power per hour. From a test by another member, a steam consumption of 306 lb. per useful horse power per hour was given by a pump discharging 70,000 gallons per hour. But from the result (also presented during the discussion) of a trial made by Professor T. H. Beare, on behalf of the Pulsometer Company, with a pulsometer fitted with the Grel controlling valve, a consumption of but 148 lb. of steam per horse-power hour was obtained, or 13,415 foot-pounds of work done per pound of steam consumed. The conditions under which the test was made were described as follows: An unlagged vertical boiler standing in an open yard, and an unlagged steam pipe 62 ft. long to the pulsometer; mean boiler pressure 55 lb. per square inch, feed water supplied 361 lb. per hour; measured height of lift 73 ft., or by pressure gauge 84·4 ft., including friction. The water pumped was 5,738 gallons per hour, which, against the head of 84·4 ft., represented 2·45 horse power.

The results given above should be compared with the particulars we have previously given concerning the steam consumption of pumps of the boiler-feed type. It will be found that the pulsometer working expansively, as described, compares very favourably with many of such pumps.

CHAPTER X.

POWER PUMPS.

THE operation of every pump is effected by the application of a power or force of some kind, but the term "power pump" is employed both in this country and in America as descriptive of a pumping machine which is driven through belting or through gearing, or both. We have previously considered self-contained pumps, or pumping engines as they should properly be termed, in which the motor or motive power machine is formed integrally with the pump itself. But a power pump receives motion only by transmission from an independent motor, which may also be giving out power for several other purposes.

The simplest power pump, and the most quiet in working, is made by connecting the pump rod or rods with a crank shaft which is driven by a belt without the intervention of gear wheels. The working of a geared pump is always attended with noisy rumbling, and if the gear wheels have their teeth cast with them, and are of the type generally turned out from an ordinary foundry, there will also be intermittent knocking so disturbing as to make the pump altogether inadmissible for some services. With accurately cut gears—ensuring effectual contact of the engaging teeth throughout their entire width, and thus permitting of the employment of a fine pitch—the noise is much lessened. The improvements effected in recent years in wheel cutting machines, and machine tools generally, have made it possible for makers to employ wheels of considerable size, having teeth cut from the solid metal, without putting a prohibitive price on their pumps.

The practical incompressibility of water is exemplified in a most unpleasant manner in the working of a power pump fitted with wheels having teeth which indifferently gear with each other, and have appreciable "back lash." Fig. 61 is a sectional sketch diagram representing a geared power pump. On the out-stroke of the water piston A (or its movement

in the direction indicated by the arrow 1) the water pressure will act upon the front end of the piston, and the connecting rod B will be in tension. As the connecting rod approaches and passes over the "dead centre" C the piston will be brought to rest, and its out-stroke thus completed. But as the connecting rod moves away from its dead centre the piston B, on commencing its return stroke, impinges upon the water at the back of it. The pressure is thus suddenly transferred from the front to the back of the piston, and the connecting rod is relieved from the tensile and subjected to a compressive stress. There is no cushioning effect; the piston is as suddenly obstructed as though it had impinged upon an armour plate. A similar water hammer action

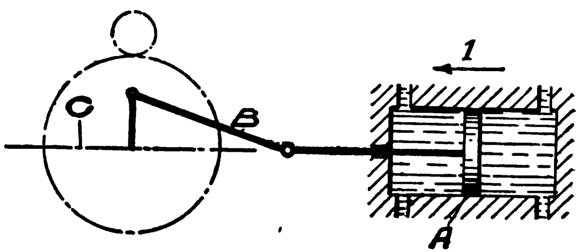


FIG. 61.

being set up at the end of every stroke, or twice in each revolution, it will be readily understood that a violent knocking must occur if the pump gear wheels have much "back lash" between the engaging teeth.

For pumping against heavy pressures, and for dealing with large quantities of water, the use of gearing in a power pump is generally unavoidable, but unless the setting up of a noise rivalling that of a steam hammer is of no moment, every care must be taken to ensure that the wheels shall gear in a very efficient manner.

POWER REQUIRED.

The theoretical horse power required to raise a given quantity of water in a certain time, through a stated height, is simply the product of such quantity in pounds delivered

per minute and the height in feet, divided by the number of foot-pound units representing 1 horse power, viz., 33,000. Thus the theoretical horse power required to raise 100 gallons (1,000 lb.) of water per minute through a height of 100 ft., is—

$$\frac{1000 \times 100}{33000} = 3 \cdot 03 \text{ H.P.}$$

It should be remembered, in referring to the catalogues of pumping machinery by both British and American makers, in which tables are frequently given, that whereas a British imperial gallon has a capacity of 0·16 cubic feet, and contains 10 lb. of water, the United States gallon has a capacity of but ·133 cubic foot, and contains only 8·33 lb. of water. Thus the theoretical power required to raise 100 United States gallons per minute through a height of 100 ft. will be but 2·50, as against the 3·03 horse power for 100 British gallons.

But in providing for the working of a power pump, we must remember that the friction of the moving parts of the pump itself, of the gearing, and of the water through the pipes will absorb a considerable amount of power. In some cases the power so absorbed may equal the power required to raise the water—in other words, the mechanical efficiency of the pump (or the ratio between the actual work obtained from the pump and the amount of power applied to it) may be as low as 50 per cent.

In calculating the theoretical power, it is, of course, the full height of lift that must be taken, or the height from the surface of water to be pumped to the highest point in the rising main. The height of the suction lift of the pump must not be deducted.

Size of Belting.

A power pump, as sent from the makers, may or may not be provided with a belt or driving pulley. When a pulley is provided the makers are usually careful to employ one of such a width of face as will receive a belt of ample strength to transmit the necessary power. Good leather belting will safely withstand a working stress of 70 lb. per inch width of

single belt. The ratio between the tension on the tight side of a flat belt and the pull transmitted thereby to the rim of a pulley will depend on the length of the arc of contact between the belt and the pulleys around which it passes. The greater the ratio between the diameters of a pair of pulleys, and the nearer they are together, the less will be the arc of contact between the belt and the smaller pulley, which is the one that must be considered. With equal pulleys the arc of contact on each will be the semi-circumference, and the ratio between the tension on the tight side of belt and the pull transmitted to the pulley will then be 5 : 3, so that with 70 lb. tension per inch width on the tight side of belt the pull transmitted to the pulley rim will be

$$70 \times \tfrac{3}{5} = 42 \text{ lb.}$$

With double belting we may transmit a pull of just double the intensity permissible with a single belt, or 84 lb. at the pulley rim, as against 42 per inch width of single belt.

The width of belting required can be readily calculated when we know the rim speed at which the driving pulley must be run and the rate at which the water is to be pumped. As an example, let us assume that we have a power pump with a $6\frac{1}{2}$ in. double-acting ram or water piston, and a stroke of 12 in., which is supplied by a maker for a delivery of 5,000 gallons per hour, against a total head of 240 ft., when the crank shaft is run at 35 revolutions per minute. If we know that the belt pulley is 30 in. diameter, and that there is a set of single purchase gearing (or a single reduction of gearing, as the Americans term it), having a ratio of, say, 5 : 1, we can at once calculate the rim speed of pulley or the speed of the belt as follows:

$$\left.\begin{array}{l}\text{Speed of belt}\\ \text{or pulley rim}\end{array}\right\} \begin{array}{l} = 35 \times 5 \times 2\frac{1}{2} \times 3\frac{1}{7}, \\ = 1375 \text{ ft. per minute.}\end{array}$$

$3\frac{1}{7}$ represents the ratio between the circumference and the diameter of the pulley.

Now, the amount of work required to raise the water is—

$$5000 \times 10 \times 240 = 12{,}000{,}000 \text{ foot-pounds per hour.}$$

Taking the efficiency of the pump to be two-thirds, or 66·66 per cent, then the actual expenditure of work per hour must be—

$$12000000 + \frac{12000000}{2} = 18000000 \text{ foot-pounds.}$$

The speed of the belt and of the pulley rim is, as we have seen, 1,375 ft. per minute, and therefore the pull required at rim of pulley will be—

$$\frac{18000000}{1375 \times 60} = 218 \text{ lb.}$$

Thus, in this case, the width of single belting required is—

$$\frac{218}{42} = 5 \text{ in.}$$

Such a width will give ample strength; if care is taken to see that the belt is run with its lower side as the tight or driving side, and there is a considerable horizontal distance between the pulleys, then, as the belt may be worked with but a moderate initial tension, the arc of contact with equal pulleys will exceed the semi-circumference, and a narrower belt may be employed.

TYPES OF POWER PUMPS.

Fig. 62 represents a vertical type double-ram power pump, by Messrs. Frank Pearn and Co. Limited. of West Gorton, Manchester. The pump has two single-acting rams, connected to a crank shaft, which is driven by belting without the intervention of spur gearing. The makers recommend it for lifts up to 80 ft. vertical.

Fig. 63 illustrates a vertical treble-barrel geared ram pump, by Messrs. Joseph Evans and Sons, of Wolverhampton, suitable for heads up to 300 ft. vertical. As will be seen, the pump is kept very compact by the use of short connecting rods and crosshead guides for the three single-acting rams.

Fig. 64 represents the horizontal form of the same type of pump, also by Messrs. Evans. It is without gear, and is

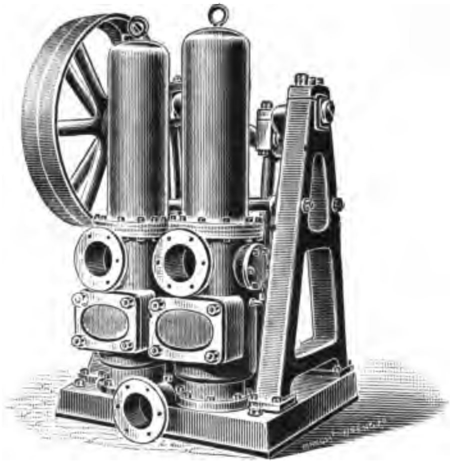


FIG. 62.

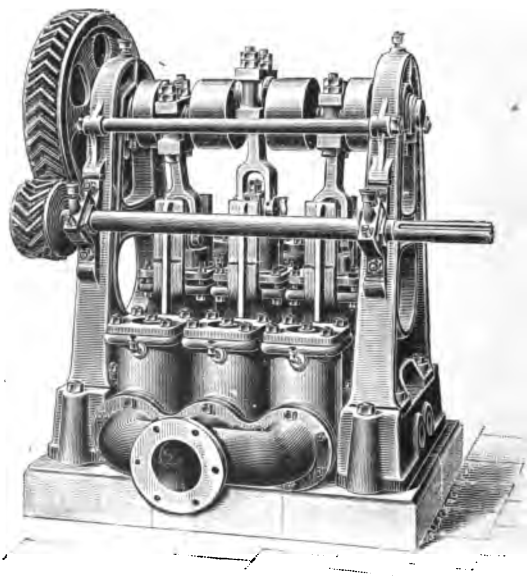


FIG. 63.

designed for either belt or wire-rope driving. Each of the three rams is provided with a crosshead working in slipper guides.

Fig. 65 illustrates a three-throw horizontal ram or plunger pump, by Messrs. Hayward, Tyler & Co., of 99, Queen Victoria Street, London, E.C., having cast-iron crossheads and slipper

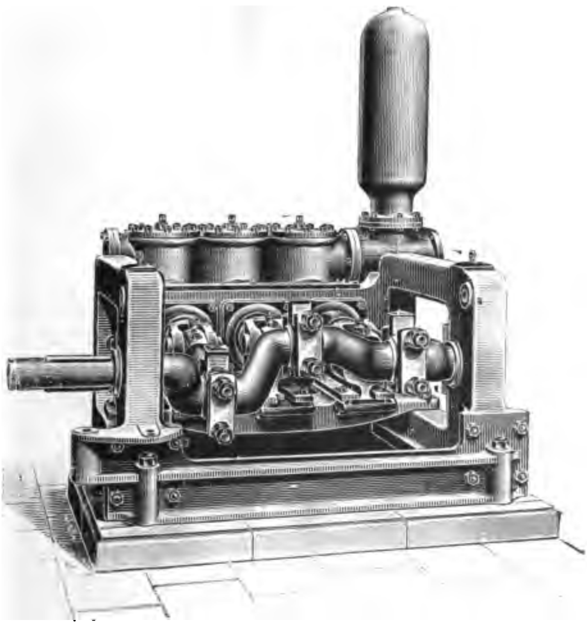


FIG. 64.

guides. The pump shown has three 15 in. plungers, with a stroke of 36 in., and is provided with two sets of gearing with sliding pinions, so that it may be driven at two speeds, for the delivery of 60,000 and 40,000 gallons per hour respectively. The belt driving pulley, 10 ft. in diameter, and weighing five tons, acts also as a flywheel. Air and vacuum vessels are attached to the delivery and suction

mains respectively; the vessels are of riveted steel plates, and are fitted with gauge or sight glasses.

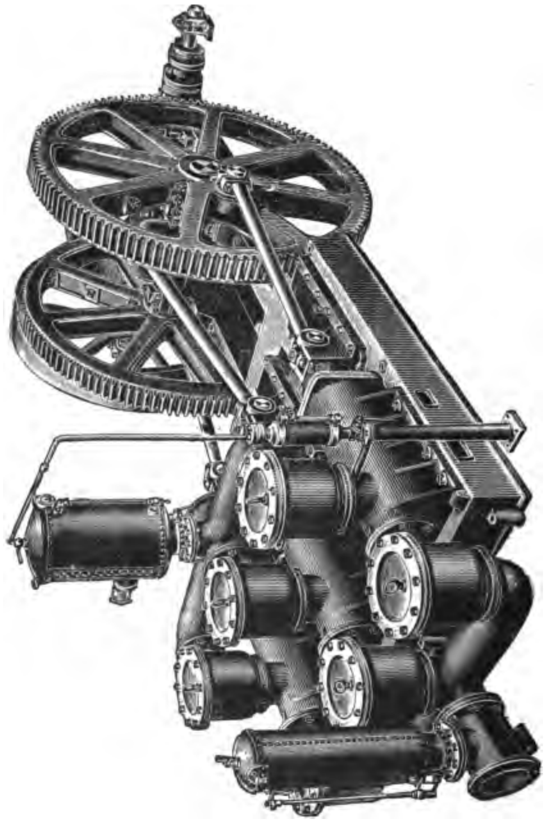


Fig. 65.

Fig. 66 is an illustration of another treble-barrel or three-throw pump, by Messrs. Hayward, Tyler, and Co., to raise 10,000 gallons per hour against 500 ft. head. The pump has slotted-steel cranks, cast-iron plungers 8 in. diameter,

cast-iron glands bushed with gun-metal, and gun-metal neck bushes. The valves are of indiarubber, on gun-metal grids, with copper stalks and gun-metal guards. The air

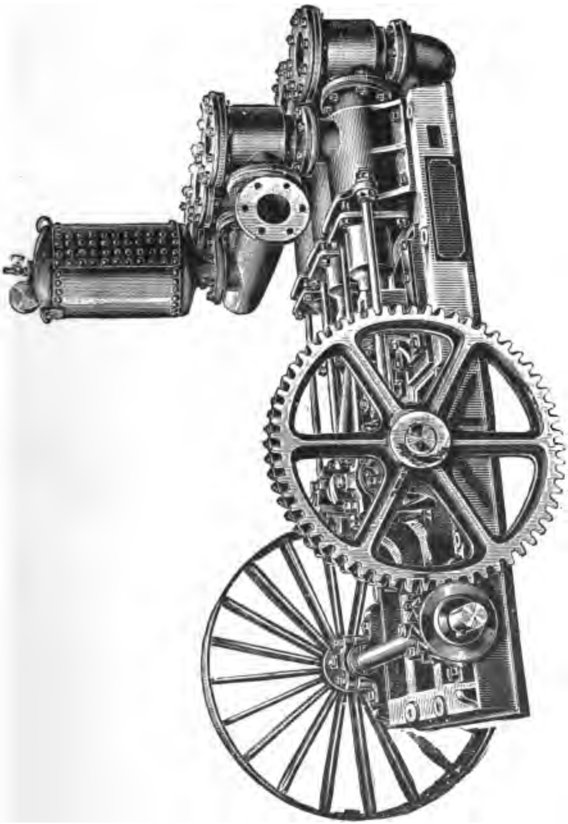


Fig. 66.

vessel is of riveted steel plate, and is fitted with gauge glass and pressure gauge. The gear wheels are of steel, with double-helical teeth, and drive the crank shaft at 40 revolu-

tions per minute. The counter shaft, or first-motion shaft, has a Rodger's wrought-iron double-arm pulley. The main plummer blocks are tied to the barrels by bright rods.

A treble-hydraulic pump, by Messrs. Hayward, Tyler, and Co., to work against a pressure of 1,200 lb. per square inch, is illustrated at fig. 67. The three plungers are each 5$\frac{1}{8}$ in. diameter by 8 in. stroke, and with a crank speed of 36 revolutions per minute deliver 70 gallons in the same time. Mitre pattern gun-metal valves are employed working on

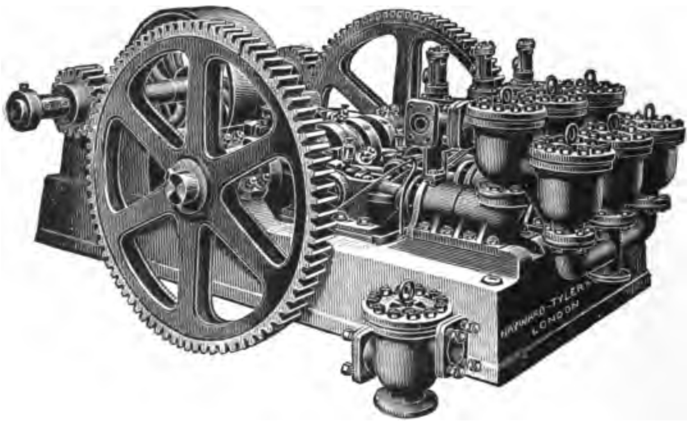


FIG. 67.

gun-metal seats. The crank shaft is of slotted steel, 6 in. diameter in the bearings. There are two machine-cut steel spur wheels and pinions, and the makers state that each pair has ample strength to transmit the whole of the required power. The width of the gear is 6 in. Spring relief valves are provided.

Fig. 68 represents a triplex-horizontal heavy-pressure boiler-feed or mine pump, by the Worthington Pump Company, of 153, Queen Victoria Street, London, provided with relief valves.

Fig. 69 is an illustration of a vertical triplex "stuff" pump by the same makers, especially built for use in paper

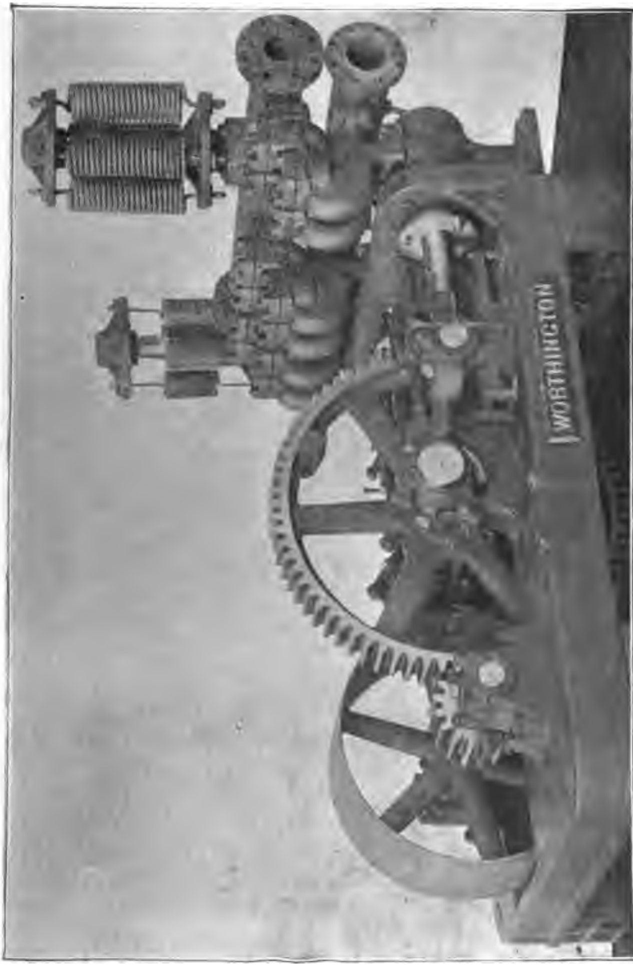


Fig. 68

and pulp mills. The makers state that the castings are all very heavy and well proportioned, that the crank shaft and connecting rods are of steel, the gears of hard-gear iron accurately cut, and the bearings of extra size and length,

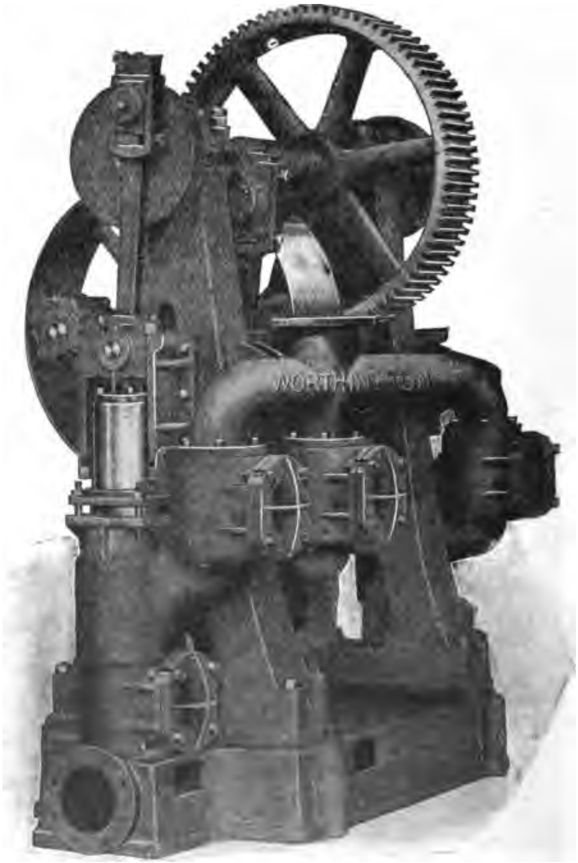


FIG. 69.

lined with best deoxidised babbitt. The connecting rods have adjustable boxes at both ends, whilst the crossheads have adjustable shoes, and move in cylindrical ways.

CHAPTER XI.

Electric Pumps.

Ordinary power pumps, such as described and illustrated in our last chapter, may be electrically driven by connection through belting or gearing with suitable motors. But the term "electric pump" is properly applicable only to the combination of a pump and its motor upon the one base or framing, or with a motor employed for no other service, just as the term "steam pump" is descriptive only of a pump combined with its own exclusive engine.

Both direct or continuous current motors and alternating-current motors may be employed for electric pumps, but motors and electric fittings that give satisfaction on other services may be quite unfitted for the work of pumping water. And as regards the pump itself, it is of extreme importance to select a type that will give a steady propulsion of the water column. A pump having but one single-acting ram or plunger is most unsuitable for operation by an electric motor, owing to the very irregular distribution of the work. A duplex-type pump having two double-acting plungers, and a triplex-type pump, are well adapted for electric service.

In some electric generating stations electrically-driven pumps are employed instead of steam pumps for feeding the boilers. It has been suggested that the primary reason for this practice is that it affords an opportunity of showing to visitors the application of electricity—the power on sale—to the pumping of water. An electric pump is not, however, so well adapted for boiler feeding as a steam pump, chiefly because the former has not that "flexibility of regulation" by virtue of which the latter can be readily set to keep that very desirable "constant water level" in the boiler under varying rates of evaporation. It is sometimes urged that a saving of fuel is effected by the use of electrically-driven as against steam pumps. Such a claim cannot, however, be sustained when the comparison is made with high-class pumps. As an example of the results that can be obtained

with well designed and constructed steam pumps, we may here mention that in a paper on marine engineering, read by Mr. James McKechnie, at the 1901 summer meeting of the Institution of Mechanical Engineers at Barrow-in-Furness, he gave results of tests which he had made on Weir direct-acting (or non-crank and flywheel) steam pumps, showing in one case a mechanical efficiency (or ratio between the work performed by the steam on the steam piston and the work given out by the water plunger or piston) of 94 per cent, and a steam consumption, in the case of a compound pump, of 31 lbs. per indicated horse power per hour, with a mechanical efficiency of 92½ per cent.

The water output of electrically-driven pumps is sometimes regulated by means of resistance coils or rheostats, but the electrical energy not required for pumping is then simply wasted by dissipation. An American writer has well described this and another system in the following terms: "The only method of controlling the speed of direct-current motors known to the average steam engineer, who as likely as not finds that in spite of his protests a new electric pump has been added to the plant of which he is in charge, is to pile extra resistance in the main circuit, and it is not unusual to see the rheostat quite as large as the motor itself. While this may answer in a few instances, it will never be an acceptable solution of the variable speed motor problem, as the speed is cut down at the expense of the efficiency, the electrical energy is expended in heat overcoming the resistance in the many extra coils of wire, and it takes just as many amperes from the dynamo to run at 1 revolution per minute pumping 1 gallon, as to run at 1,000 revolutions per minute pumping 1,000 gallons. The function of a rheostat is for stopping and starting only; that is all that should be expected of it.

"Another method, which is only a degree better than the 'resistance rheostat,' is to adopt some intricate system of interchangeable gearing or clutches between the pump and motor. Such schemes often work out very prettily on paper, but in practice they prove too complicated. They add to the first cost and bulk of the machine, and render it liable to get out of order at the critical moment."

A system of controlling the speed of electric pumps—with direct-current motors—that has given satisfaction where a variation of about 25 per cent will meet requirements, consists in the employment of motors with specially-wound field coils, so that the amount of current flowing through the field magnets can be varied (by means of a small rheostat) for the purpose of varying the speed of the armature, and consequently of the pump output. Within the aforesaid limits of variation the amount of current used is in proportion, with such motors, to the quantity of water pumped.

In another system advantage is taken of the fact that "when two electrical units are run in series the voltage or potential is divided between them, and the speed cut down one-half." Thus by employing two motors of equal horse power they can either be run in multiple, when the full available voltage will act upon each, and the pumps worked at full capacity, or in series for the purpose of reducing the voltage and therefore the speed and the output of the motor and pumps by one-half. To arrange the motors for such service nothing is required beyond a few additional wires, so that by the mere opening and closing of a switch they can be run in series or multiple at will.

But though for boiler feeding a steam pump is preferable to an electric, for the reason already given, and also because of the greater handiness of having a pump adjacent to the boiler, depending for its working only on the steam therefrom, there are other services for which electric pumps are eminently adapted. Thus, for what is known as house tank service and also for hydraulic elevator or lift service, small electrically-driven pumps may be employed where a steam pump would be altogether inadmissible. It is true that quick-running elevators or lifts in large office blocks and hotels may be with greater advantage directly worked by electricity, by connecting the car-winding mechanism with a motor, yet such a system involves the use of a much more powerful motor than is necessary to operate a small pump supplying either a gravity (or elevated) tank or a pressure tank. In the one case the motor must be capable of raising the full load at the required speed, whereas with a

hydraulic elevator or lift supplied from a tank of sufficient capacity to contain enough water for several car trips, the pump motor has simply to supply sufficient power to make up the tank water level between and during the lift trips or journeys. This system has been termed the "hydro-electric system." The plant is perfectly automatic in action, for by means of a suitable snap switch the pump is started and stopped as required by the rise and the fall of the water in the tank. When motors especially wound to run at low speeds are employed, no gearing is necessary with these small electric pumps; disturbing noise is thus effectually avoided.

An electric triplex-ram or plunger-type pump, for heavier services, as made by the Worthington Pump Company, of 153, Queen Victoria Street, London, is illustrated at fig. 70. The three single-acting rams or plungers are arranged horizontally and driven, through a single reduction of spur gears, by a direct-current motor.

The Worthington Company also construct electric fire pumps. Such pumps they provide with triplex differential rams or plungers for obtaining a very uniform delivery, and with exceptionally large valve areas and water passages to ensure the complete filling of the ram chambers when the machine is running at a high speed. These pumps can be arranged to deliver direct into fire pipe lines and sprinkler pipes, and the makers state that they can be relied upon for maintaining at all times a pressure on the entire system. The electrical device for starting the pump automatically is practically the same as that which the makers have used successfully on elevator service. They describe it as follows: "A regular rheostat is placed in the circuit, the arm of the rheostat being operated by a small hydraulic piston and a weight, instead of operating it by hand. The water pressure in the fire line, acting on the top of the piston, holds the weight up with the arm thrown back and the circuit broken. If a hydrant is open, or a sprinkler lets go, the water pressure falls, the piston is relieved of pressure by means of a regulator, and the suspended weight draws the arm slowly across the face of the rheostat, making the circuit and gradually cutting out the resistance. In this way the pump

FIG. 70.

is started automatically, the action on the motor being precisely similar to the operation of starting by hand. Any excess in the quantity of water pumped will again raise the pressure in the system and stop the pump. By merely laying a wire this same mechanism can be made to ring an electric gong placed anywhere in the factory, thus giving the alarm as soon as the first sprinkler opens."

Electric mine pumps, owing to the facility with which the current can be conducted down the pit, are finding increasing favour for services in mines equipped with modern plant throughout. With electric pumps we have no steam pipes to provide, no condensation to guard against, and no exhaust steam to deal with. As the work on the pumps is constant, that "flexibility of regulation" to which we have referred is not necessary; electric mine pumps will therefore compare the more favourably with pumps worked either by steam, or, as is sometimes the practice, by compressed air.

Fig. 71 is an illustration of a vertical electrical sinking pump as constructed by Messrs. Hayward, Tyler and Co., of 99, Queen Victoria Street, London, for the Oerlikon Machine Co., of Zurich. The makers state that these pumps have differential plungers—10½in. and 15in. diameter by 18in. stroke—driven by an enclosed crank receiving its motion through a double reduction train of double helical gearing from a motor placed at the top of the pump, as illustrated. The gearing and motor are protected by a sheet-iron case, and the whole machine is suspended by a chain and eye bolts. There is a bye-pass valve for starting, and a primary valve for the suction pipe. The pump illustrated was constructed to raise 12,800 gallons of water per hour against a head of 400 ft., the motor being 36 horse power of the Oerlikon type.

A very large electrically-driven horizontal double-acting triplex pump has been installed by the Worthington Pump Company, at Austin, Texas, concerning which they supply the following particulars: "The situation at Austin, Texas, where this pump is installed, is in many respects peculiar. The city has practically an unlimited volume of water, with a fall of 60 ft., and can afford to waste any amount necessary to raise the needed supply to the highest point required, and


HAYWARD TYLER & C°
BYE-PASS
OR STARTING VALVE
DELIVERY
PRIMING
PIPE
SUCTION.

Fig. 71.

is therefore completely independent of steam as a motive power. The water power thus secured operates a power station of about 3,000 horse power, which furnishes light and power for the entire city. Electricity generated at this station is transmitted to the pumping station, where a 300 kilowatt three-phase synchronous motor is used to drive the pumping engine. The motor armature is about 18 ft. in diameter, runs at 100 revolutions, the voltage being 2,200, and the number of cycles 72; it is coupled directly to the pump counter shaft which carries a pinion gearing with a spur wheel on the crank shaft. As the synchronous motor cannot be started up against a full load, a 72 in. friction clutch is provided on the counter shaft on the opposite side of the pinion from the motor. This is made possible by the use of a quill or hollow shaft, the brake shoes of the clutch being mounted on the counter shaft, whilst the friction wheel of the clutch and the pinion are keyed to the quill.

"The pump proper is a horizontal triplex having three centrally packed double-acting plungers 18 in. in diameter by 24 in. stroke. It has a daily capacity of 6,000,000 U.S. gallons, and is designed for a working pressure of 130 lb. to the square inch. The spur wheel is carried on a solid forged shaft with U-shaped cranks. It is 146·4 in. pitch diameter, with cast-iron body, and mortised teeth made of seasoned maple. The pinion is steel, 36·6 in. pitch diameter and 26 in. face. There is only one reduction of gears. This is made possible by the slow speed of the motor, and is a marked improvement on previous plants which have high speed motors, double, and even triple reduction of gears, with the consequent high periphery velocities, noise, and wear and tear."

CHAPTER XII.

Centrifugal Pumps.

These well-known machines, with which water is raised by the centrifugal force set up by a revolving disc or fan, were first brought prominently before the notice of engineers and pump users generally at the Great Exhibition of 1851.

A crude form of centrifugal pump was known as early as the middle of the 18th century. Rather more than half a century later, in the year 1818, a centrifugal pump was constructed at Massachusetts, U.S.A., and became known as the Massachusetts pump. It resembled an

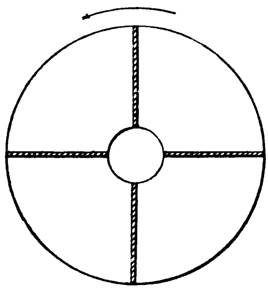


Fig. 72.

ordinary fan blower, and comprised a horizontal shaft having four straight blades enclosed within a cylindrical casing. But the pump had to be placed below the level of the water to be raised, for it is stated that "the vacuum power" was small.

At the Great Exhibition of 1851 Appold introduced his pumps having fans or impellers with curved blades or vanes (in place of vanes of the radial type shown at fig. 72), and with other improvements which enabled him to show an efficiency three times greater than could be obtained with the old machines. Appold's improvements established the position of the centrifugal pump as a most suitable machine

for raising large quantities of water against a low head or pressure.

The frictional loss in the working of a reciprocating pump is practically constant, and independent of the useful head or pressure against which the water has to be delivered.

Take the case of a reciprocating pump working against a pressure of, say, 60 lb. per square inch or 139 ft. vertical head, and requiring a pressure of, say, 15 lb. per square inch to overcome the frictional resistance. Out of the total energy supplied to the machine $\frac{15}{75}$ths, or 20 per cent, will be absorbed in friction; the remaining 80 per cent represents the useful work done, or the mechanical efficiency of the machine. But if the pump is set to work against a head of but 10 lb. per square inch, the frictional resistance will still require a pressure of 15 lb. to overcome it, and thus out of the total energy supplied only $\frac{10}{25}$ths, or 40 per cent, will be accounted for in useful work done, the remaining 60 per cent being absorbed by friction.

It will thus be seen that "the efficiency of reciprocating pumps diminishes with the lift." With centrifugal pumps there is no such diminution; indeed, it has been stated, though not perhaps with strict accuracy, that they work to best advantage on services where the lift in feet can be represented by a unit figure.

An increase in the head against which the water has to be delivered by a centrifugal pump necessitates an increase in the speed of the revolving disc, fan, or impeller. The theoretical ratio between the head of water and the speed of the pump, when the revolving disc of the latter has arms or vanes which are radial at the periphery, is expressed by the formula—

$$h = \frac{V^2}{g}$$

where h = head in feet (measured from the surface of the water to be raised);
V = velocity of periphery of pump disc or fan in feet per second;
g = accelerating force of gravity, say 32·2.

As the height through which any moving body must fall freely to attain a given velocity is expressed by the well-known formula—

$$h = \frac{v^2}{2g}$$

(of which we made use in an earlier article dealing with pipe areas), it might be erroneously concluded that the same formula would also apply here and that V represents also the velocity of the water. But on reflection it will be seen that the water in passing through the centrifugal pump receives energy which tends to propel it in two directions, viz., in an outward direction along the radial arms, and also in a direction tangential to such arms or blades of the fan, disc, or impeller. At the instant of reaching the periphery of the impeller the velocity of the water in each direction will equal the peripheral velocity, so that the water will be discharged from the periphery in a direction which is the resultant of the two directions aforesaid, and its velocity will also be a resultant of what we may term the two component velocities. Thus it is that

$$h = \frac{2V^2}{2g} = \frac{V^2}{g}.$$

As an example let us suppose that the rim velocity of a pump disc with radial arms is 36 ft. per second. The maximum theoretical head against which the water can be delivered with the pump running at the given speed will then be

$$\frac{36 \times 36}{32 \cdot 2} = 40 \cdot 2 \text{ ft.}$$

But the actual head against which the pump can deliver water will be considerably less than 40 ft., depending upon the care taken in the design and construction of the pump to prevent sudden changes in the direction and velocity of the water. With the old style of radial arms, as shown at fig. 72, there is much loss through useless churning of the water. It is, of course, also of great importance that the bearings upon which the fan shaft rotates and the means

employed for driving the fan or disc be mechanically efficient.

To utilise the energy which might otherwise be spent in the formation of eddies in the discharge pipe of a centrifugal pump, Professor James Thompson suggested, subsequent to the introduction of the Appold improvements, the use of what is termed the "diffusor" or "whirlpool chamber." Such chamber surrounds the revolving disc, and the water delivered into it from the latter is allowed to continue its whirl or rotation for effecting a more gradual and efficient conversion of the kinetic energy into the pressure energy required to force the water up through the delivery pipe.

But in order to get full benefit from a diffusor or whirlpool chamber it would in most cases require to be so large as to cause more inconvenience than would be compensated for by the practical advantage obtained. Moreover, as the vanes of the disc are in British practice curved back, and not made of the radial form previously referred to, the velocity of the water on leaving the fan, and consequently its kinetic energy, is diminished; a further reduction of the kinetic energy by its conversion into pressure energy before the water leaves the disc, impeller, or fan, is also effected in some centrifugal pumps by increasing the area of the water space towards the periphery of the disc. The whirlpool chamber or diffusor may therefore be greatly reduced in dimensions, and in some cases entirely dispensed with, in the construction of the pump, without loss of efficiency in its working.

From the whirlpool chamber, or directly from the periphery of the disc, the water passes into the volute or spiral chamber, having a section uniformly increasing as it approaches the discharge or delivery pipe to receive the increasing volume of water from the disc. Conical suction and discharge pipes are sometimes employed with advantage.

Figs. 73 to 76 represent four types of discs or impellers, as adopted by different makers. Figs. 73, 74, and 75 show (in side elevation) three curved or sloping vane types by three different English makers, whilst fig. 76 represents (in plan) the modified radial vanes of a horizontal type centrifugal pump by French makers.

Unless a centrifugal pump can be fixed below the surface of the water to be raised, it is necessary to charge the casing

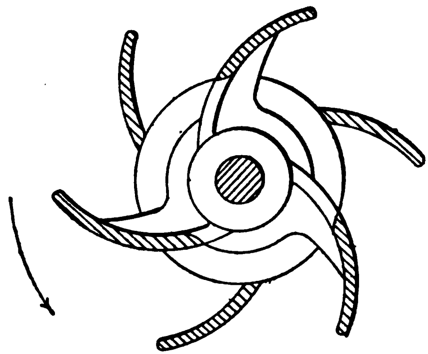


Fig. 74.

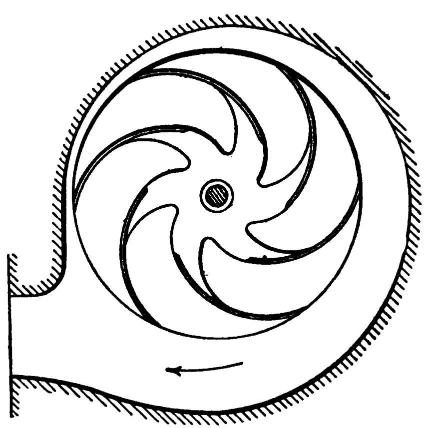


Fig. 73.

with water before starting. For this purpose a steam ejector can be employed to exhaust the air from the pump casing and pipes.

Fig. 77 is an illustration of a centrifugal pump, for belt driving, by Messrs. Drysdale and Co., of Bon-Accord Works,

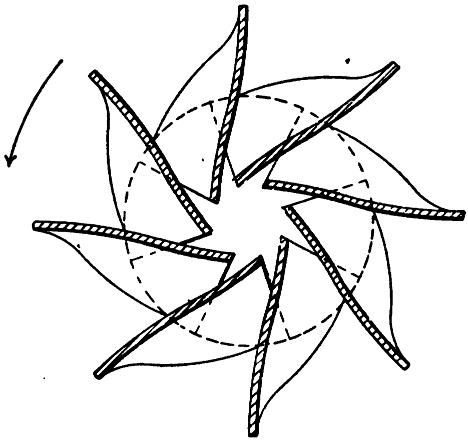


FIG. 76.

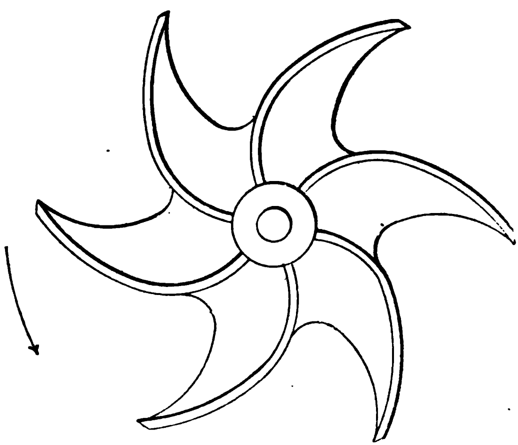


FIG. 75.

Glasgow. The makers state that "while these pumps will meet the requirements of all users for ordinary low lifts,

they will be found specially advantageous for irrigation, drainage, and similar work. The impellers or discs are specially wide to permit of the passage of any solid matter likely to be allowed inside, and a cleaning door is provided to meet the case of any special obstruction." In their lists, published with the pump as illustrated, the makers give the speeds necessary for lifts of 10 ft. and 18 ft. respectively. Thus a small pump, to deliver from 60 to 80 gallons per minute, must be run at about 1,000 revolutions per minute

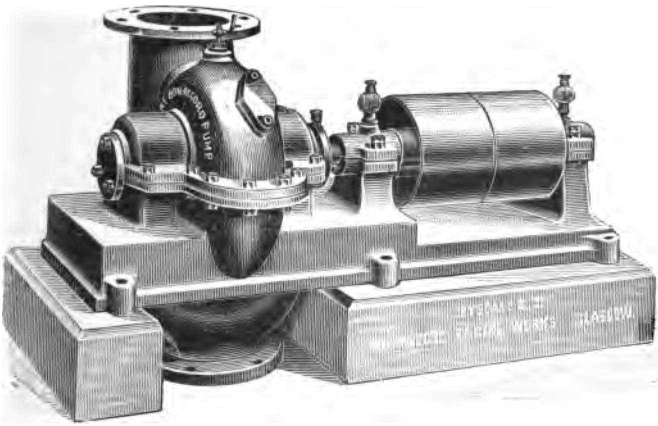


FIG. 77.

when working against a 10 ft. head, and 1,350 revolutions for a head of 18 ft. A larger pump, to deliver from 2,200 to 2,700 gallons per minute, is run at 345 revolutions per minute when working against a head of 10 ft., and 490 revolutions when working against an 18 ft. head. Pumps designed for high lifts (from 25 ft. to 50 ft. or more) are provided with fans of larger diameter than the low-lift types, in order to obtain the required peripheral velocity of the impellers or discs without driving the same at such a high rate of rotation as to cause inconvenience.

Fig. 78 is an illustration of a combined centrifugal pump and electric motor, also by Messrs. Drysdale and Co.; whilst

fig. 79 represents a combined pump and vertical steam engine by the same makers, as employed for the circulation

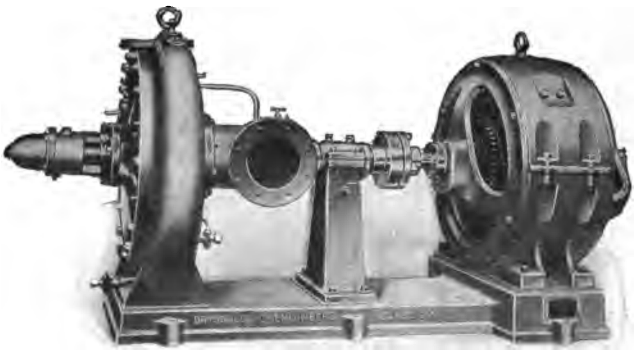


Fig. 78.

Fig. 79.

of surface condensing water in the mercantile marine and for other services.

Messrs. Gwynnes Limited, of Hammersmith Works, London, W., who have manufactured such machinery for half a century, give the following terse summary of some advantages possessed by a centrifugal pump :—

"It can be erected with ease and celerity.

"It works with an easy rotary motion, without valves, eccentrics, or other contrivances which consume power in friction.

"It is economical in use, simple in construction, and of very great durability.

"It discharges a continuous and steady stream without air vessels.

"It is little affected by sand, mud, grit, or other foreign matter in the water; in the larger sizes it will admit the passage of solid bodies 6 in. diameter, and the smaller sizes in proportion, without injury."

As compared with the full theoretical efficiency of 100 per cent, Messrs. Gwynnes give the average efficiency of their centrifugal pumps as 75 per cent, and claim that so far back as 1862 a centrifugal pump of their manufacture gave, at the International Exhibition of that year, an efficiency of 83 per cent.

As an example of the very accurate balancing of their machines, Messrs. Gwynnes instance the experimental running with an empty pump of one of their centrifugal pumping engines, or centrifugal pump coupled direct to a vertical steam engine, at 550 revolutions per minute, when not fastened down but merely resting on timbers. The makers state that no vibration was experienced during such test.

For emptying graving docks, and for other services where enormous quantities of water have to be rapidly discharged against a low head, the centrifugal pump is unrivalled. Messrs. Gwynnes construct centrifugal pumps for such services with discharge pipes or branches as large as five feet in diameter, and which on official trial have delivered over 82,000 gallons or 366 tons of water per minute.

Fig. 80 represents one of Messrs. Gwynnes centrifugal pumps coupled direct to one of their enclosed silent high speed engines.

Messrs. Gwynnes Limited give the following particulars respecting their combined centrifugal pumping engine, as employed with surface condensers and for similar

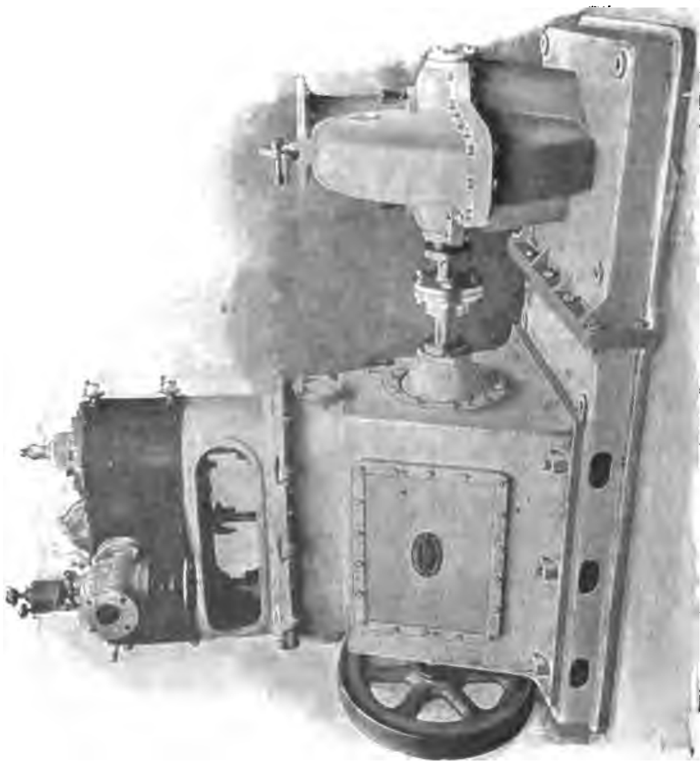


Fig. 80.

services, consisting of a vertical engine of the ordinary or open type having its crank shaft directly coupled to the spindle of the pump which is mounted on the engine bed :—

"The pump disc is of gun-metal and the spindle of steel, coated with gun-metal. All the bearings are of manganese bronze and have large wearing surfaces carefully adjusted. In many cases, especially for the Admiralty, the pump casing,

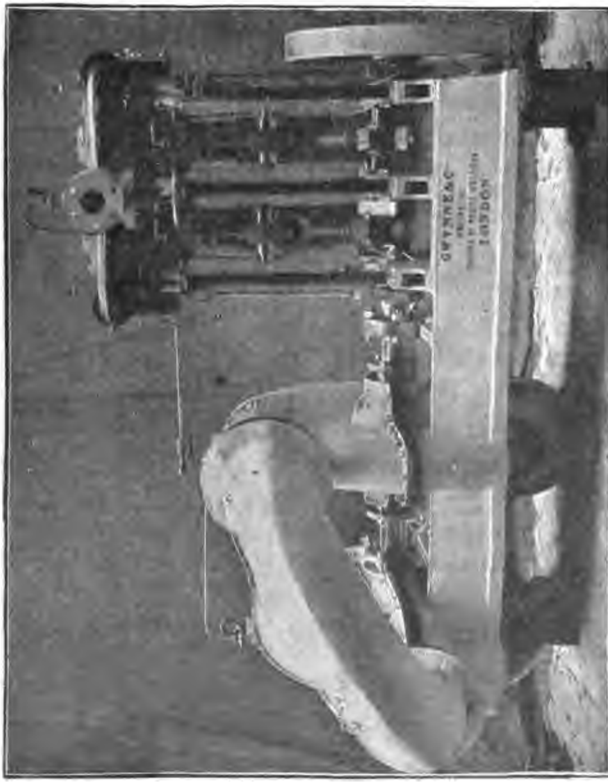


Fig. 81.

the disc, and the spindle are all of gun-metal. The pump disc and spindle can be removed and replaced without disturbing any pipe joint."

Fig. 81 represents a centrifugal pump plant for high lifts,

by the same makers. A pair of pumps arranged in series are driven direct by a double-cylinder steam engine. The delivery from the one pump flows to the suction inlet of the other pump, which imparts more energy to the water to enable it to overcome a greater delivery head.

This method of imparting the required energy to the water by means of two or more moderately sized pumps, arranged in series and rotating at the same speed, rather than by a single pump very large in diameter and running at a high velocity, may be sometimes adopted with advantage. In general, however, a reciprocating type pump would be employed for lifts above the economical capacity of ordinary centrifugal pumps.

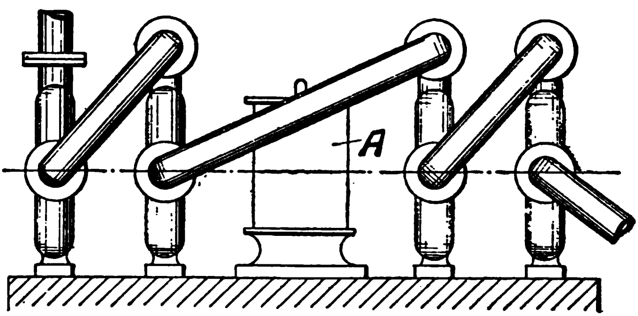


FIG. 82.

Fig. 82 is a sketch diagram representing an arrangement, recorded by *Cassier's Magazine*, in which four pumps are coupled by a French maker (whose name is not given) direct to an electric motor A, all being arranged on the one bed plate, as indicated. It is stated that the plant was employed to raise water through a height of about 157 ft. No statement is made as to the efficiency of the arrangement, but in view of the many necessary changes in the direction of flow of the water, in passing from the suction pipe of the first pump to the discharge pipe of the last pump of the series, no great efficiency could be anticipated.

Fig. 83 is an illustration of a centrifugal pumping engine for circulating purposes by Messrs. W. H. Allen, Son and

Co. Limited, of Queen's Engineering Works, Bedford. The makers state that their standard "Conqueror" centrifugal pump "is of the double suction type, the casing being divided horizontally. It is fitted with a disc of the shrouded type which is the strongest and most efficient design, and as there is no possibility of the vanes getting broken or wearing at the sides, these pumps maintain their efficiency

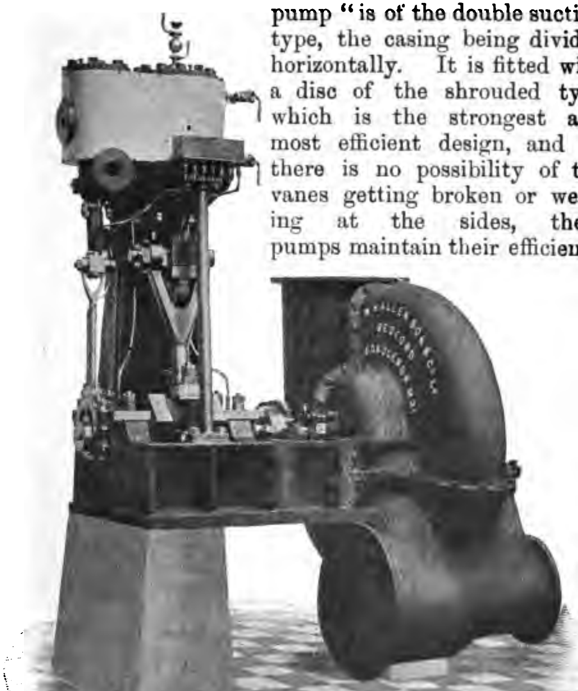


FIG. 83.

for very long periods. The spindle runs in two white metal bearings constructed in halves for ready adjustment and mounted as close as possible to the disc, thus ensuring perfectly steady running. These bearings are entirely independent of the gland." The example illustrated at Fig. 83 is of the type designed to meet British Admiralty requirements. The engine is of the single cylinder open type. The pump casing, disc and spindle are of gun metal.

Fig. 84 illustrates another example of centrifugal pumping machinery by Messrs. Allen, comprising a two-cylinder vertical double-acting enclosed engine in direct driving connection with the pump, and having extended feet cast

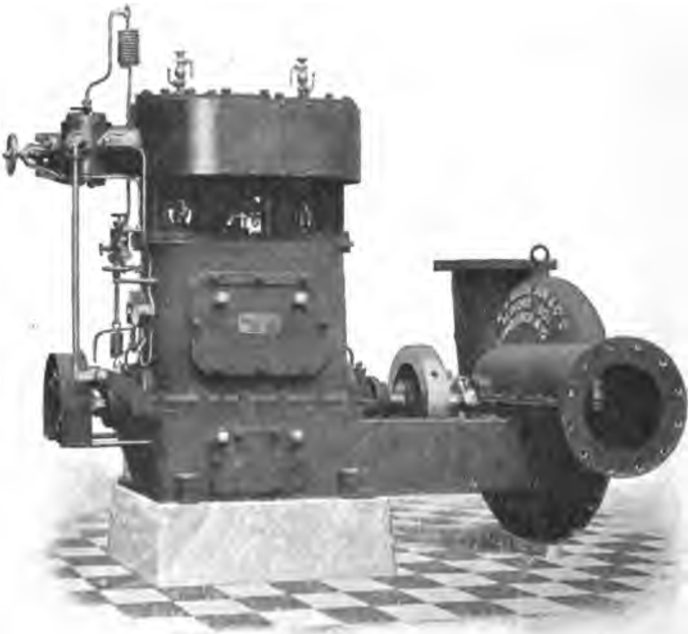


FIG. 84.

with the bed-plate for connection with the extended foot of the pump casing. The engine is fitted with a system of forced lubrication.

In connection with their belt-driven centrifugal pumps, Messrs. Allen give the following particulars concerning one of large dimensions, as made for the irrigation of a large cotton plantation in Egypt:—

"The discharge pipe is 36 in. diameter, and the pump is capable of throwing 90 tons of water per minute. The engine for driving it is a 130 horse power, passed through a treble leather belt, 21 in. wide and $\frac{5}{8}$ in. thick. The pump is made in two pieces, and every facility is provided for getting at the interior without difficulty. The pulley is made extra large, so that there may be no slip on the belt; the shaft is supported by two bearings bolted on to a massive bed plate. To charge the pump, a patent ejector is fitted on the top of casing, and the pump is also provided with a gauge glass to enable the position of the water to be ascertained when starting the pump."

For marine salvage purposes Messrs. Allen construct centrifugal pumping engines having the whole of the working parts of solid manganese bronze forgings. The following description of such engines is from the pages of *Engineering*:—

"In these pumping engines the parts usually made of steel—that is, the piston rod, connecting rod, crosshead, crank shaft, pump spindle, eccentric rod, eccentric strap, and valves, as well as the bolts and nuts,—are of manganese bronze. The engines have been designed to prevent the loss of time which frequently occurs in raising ships which are only partially submerged at low tide. In such cases the machinery has sometimes to remain under water for several days together, until advantage can be taken of a low tide to pump out the ship. With steel working parts, great difficulty arises from the journals of the shaft becoming oxidised to such an extent as to cause a quantity of minute particles of steel rust to remain in the bearings. Immediately the engines are started, seizing takes place, and then a complete overhaul has to be made, during which time the opportunity is slipping away, and when matters are put right the rise of the tide stops the work. Great difficulty was experienced in obtaining a suitable metal to form the bearings for forged bronze to work in, and many alloys were

tried before one was found which fully answered the requirements. The alloy now used is a hard mixture which runs at high speeds without heating, and wears in a short time to a smooth and glassy surface. The strength of the forged bronze is about that of mild steel (29 to 30 tons per square inch), so that nothing is lost by the adoption of the new metal."

Concerning the method of charging centrifugal pumps by exhausting the air Messrs. Allen give the following description:—

"Where the pumping engine is placed above the water it is first necessary to charge it before working. For this purpose we employ a patent ejector, which will exhaust the air and draw the water up from a depth of 25 ft. The arrangement is very simple, and yet perfect, the ejector being the smallest and most convenient contrivance that can possibly be devised for this work. It is screwed into the highest part of the pump, and also connected by a separate steam pipe to the upper part of the steam stop valve on the engine or boiler. In a few minutes after turning on steam the pump will be charged, the engine remaining stationary meanwhile. To prevent the air returning through the discharge pipe a flap valve is fitted on to end of the delivery pipe. For marine engine purposes the ordinary Kingston valve answers the purpose. For charging the large pumps we strongly recommend this method of flap valve and ejector as in every way most convenient and suitable, being much less costly and more efficient than any other means. It does away with the necessity of a foot valve, and is a much neater arrangement than an air pump or mechanical exhauster, as it is always ready, and cannot get out of order. There are a number of instances, however, where the foot valve is indispensable, and where it is as useful and convenient as the ejector."

When the water can be drawn from and discharged to the same level, it is advisable to make use of the "syphoning action," which can be done by continuing the discharge pipe below the level of the water. The pump then has only to deal with the head created by the friction of the pipes, so that in some cases the power required may be considerably reduced. It is essential that the discharge pipe be always drowned."

Some centrifugal pump makers insert in their published lists particulars as to the alleged horse power required to raise a given quantity of water per foot of height. It would be of very great advantage if the actual horse power necessary to drive the pump were stated, but if the figures simply represent work performed in raising the given weight of water through the height named, with no allowance whatever for frictional losses, they are worse than useless. A buyer who relied upon such figures in arranging for the power to drive the pump would be grossly deceived. Thus in one catalogue we find it stated that with a given centrifugal pump one horse power is required to raise 3,350 gallons of water in one minute through a height of one foot. Now, as 33,500 units of work must be performed in the raising of 3,350 gallons of water through a height of one foot, and as the rate of work represented by one horse power is only 33,000 units per minute, it will be seen that the statement is utterly misleading. If we allow that the efficiency of the particular pump referred to is 75 per cent (and we doubt whether the makers would give a guarantee to that effect), the actual horse power required to drive it on the service named will be—

$$\frac{33500}{33000} + \frac{1}{3}\left(\frac{33500}{33000}\right) = 1{\cdot}35 \text{ H.P.}$$

If the purchaser provides $1\frac{1}{2}$ horse power, he will probably find he has none to spare.

Rotary Pumps.

Though a centrifugal pump might be classed as a rotary pump the term is usually applied only to those water-raising machines which are provided with a rotating piston, which imparts a direct pressure to the water, thereby causing the latter to move in advance of, but in the same direction of motion as the piston. With a centrifugal pump the revolving disc sets up a centrifugal action whereby, as we have seen, the water is caused to flow from the centre to the periphery of the disc, and from thence into the surrounding chamber communicating with the discharge outlet.

Many forms of rotary or rotating piston pumps have been suggested and constructed in this country and in America, but hitherto they have not been extensively adopted for the varied services on which reciprocating pumps are employed.

Fig. 85 is a sectional view of the type of rotary pump known as the "Drum" pump, constructed by the Drum Engineering Co., of 33, Brook Street, Bradford. The makers supply the following description:—

FIG. 85.

"Pumps may be divided into two classes, viz., the centrifugal and the direct-acting piston pump. The former has large capacity, but is limited in power; the latter is the reverse. It has great power, but is limited in capacity.

The Drum pump combines the advantages of both without the disadvantages of either. It has the capacity of the centrifugal, with the power of the ram. It consists of a revolving piston, sweeping out the cylinder every revolution—the revolving piston dipping into a revolving valve or cylindrical drum, the openings in which are so arranged

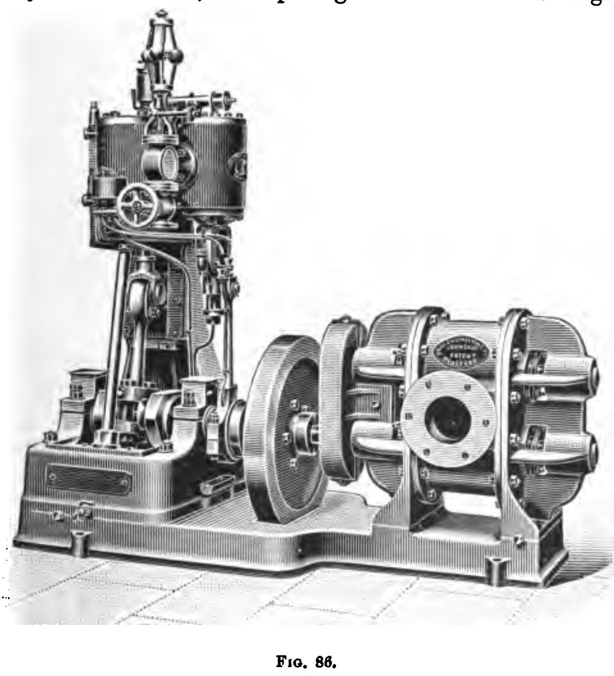


FIG. 86.

that the piston passes through without slip, back pressure, or undue friction. When the revolving piston moves round from the revolving valve a vacuum is formed into which the water flows, and is forced in the front face of the piston.

"The 'Drum' rotary piston pump differs entirely from the ordinary rotary pump. The latter, like the direct-acting piston pump, is more or less intermittent in its

action, and breaks up the flow of the fluid; the 'Drum, on the contrary, passes the water through in one continuous flow without interruption, and utilises the power in the momentum of the moving column. Thus the great advantage of centrifugal action is obtained without the cost of additional power. Another great difference between the 'Drum' and the ordinary rotary pump is that the latter has usually two revolving working pistons, one of which is driven through the gear wheels. The 'Drum' having only one rotary working piston, the friction and strain on the gear wheels and bearings of driving a second revolving piston is avoided."

Fig. 86 is an illustration showing the "Drum" pump combined with a vertical steam engine.

CHAPTER XIII.

Reciprocating Pumping Engines of Large Capacity for Waterworks, Mines, and other Services.

In the selection of a reciprocating pumping engine for a waterworks, mine, or other service necessitating the expenditure of considerable mechanical power, a purchaser must perforce study most carefully the cost of providing such power.

If tenders are invited from various makers for a steam pumping engine to deliver, say, one million gallons of water per day of 10 hours, against 100 ft. head, the proposal of one firm will probably involve an initial outlay two or three times greater than that of some other firm. But if the "duty" of the pumping plant (including engine and boiler) of higher first cost is 150,000,000 foot-pounds per 112 lb. of coal, whilst the duty of the other plant is only 50,000,000 foot-pounds, the consumption of coal per week of six days for the former would be two tons, and for the latter six tons. Thus, with its annual saving of more than 200 tons of coal, the pump of higher first cost might prove by far the better bargain.

A high duty may, however, be purchased too dearly. Pump service is severe on machinery; and mechanism, or parts of mechanism, which through intricate combination with other elements must be of delicate construction, is likely to require frequent renewal.

In considering the claims made on behalf of any particular type of pumping machinery, we have, then, to examine for economy both in power and in upkeep. And in comparing the records of tests on various engines, care must be taken to ascertain as to how far the conditions are in agreement. Thus one test may be taken on fuel of high quality, and another with but the fuel of the district, however poor it may be. One result may show the steam or coal consumption per indicated or engine horse power, and another the consumption per actual or pump horse power. In one case a pumping engine may be working against a low head, causing it to show a low mechanical efficiency as compared with an engine working against a much greater head or pressure. These and other differences must be carefully noted and allowed for, or the data may lead to a wrong conclusion.

ROTATIVE OR CRANK AND FLYWHEEL PUMPING ENGINES.

Fig. 87 is an end elevation representing one of a pair of triple-expansion pumping engines supplied in the year 1899 by Messrs. Hathorn, Davey, and Co., of Leeds, to the Corporation Waterworks of that city. One of these engines was tested by Professor W. Cawthorne Unwin, F.R.S., under a twelve hours trial on November 11th, 1899. The following notes are from his report of the trial:—

"The engine is a triple-expansion, vertical, three-crank fly-wheel engine. It has a surface condenser, the circulating water in which is the water pumped by the engine. The crank sequence is intermediate, high, low pressure. The steam valves on the cylinders are Corliss valves in the cylinder heads, with a very simple and satisfactory trip gear. The cylinder clearances are exceptionally small. The trip gear of the high-pressure cylinder is controlled by a speed governor. The trip gears of the other cylinders are ordinarily set to a fixed cut-off, but this is variable by hand

adjustment. According to the scales marked on the gear, the cut-off was at 0·26 in the low-pressure

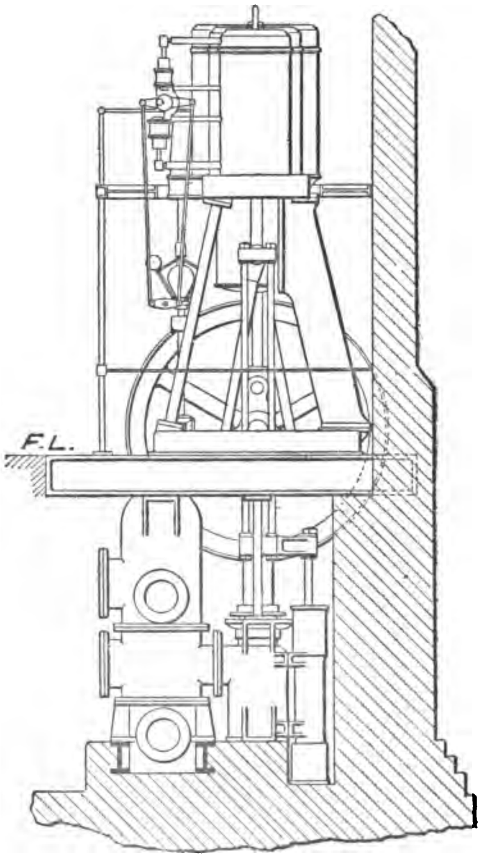


FIG. 87.

cylinder, at 0·28 in the intermediate cylinder, and at about 0·28 in the high-pressure cylinder. The steam

cylinders are steam jacketed—the high-pressure and intermediate-pressure with boiler steam, and the low-pressure with steam of about 50 lb. pressure per square inch. There are jacketed receivers between the intermediate-pressure and the low-pressure cylinders. The drainage from the jackets is led direct to the boilers. The engine has three single-acting pump rams, $13\frac{1}{2}$ in. diameter, worked from the respective crossheads, with numerous metal valves of small diameter, faced with rubber."

The contract conditions specified that the engine under full load should pump $1\frac{1}{3}$ million gallons in twelve hours with a steam consumption not exceeding 16 lb. per pump horse power. It was further specified that the delivery be calculated from the pump displacement, without deduction for slip, the head determined by tested gauges on the suction and delivery pipes, and the steam consumption calculated by measuring the condensed steam discharged by the condenser and the jacket drainage.

Leading Dimensions and Particulars.

Diameter of cylinders, 15 in., 25 in., and 40 in.
One piston rod to each cylinder, $3\frac{3}{4}$ in. diameter.
Stroke, 36 in.
Revolutions per minute during trial, 34·64.
Piston speed, 207·84 ft. per minute.
Head on pumps, 286·9 ft.
Mean steam pressure in boilers, 138 lb. per square inch.
Mean steam pressure in valve chest, 136 lb. per square inch.
Mean absolute pressure in valve chest, 150 lb. per square inch.
Mean vacuum, 27·79 in.

Results of Trial.

Water pumped per hour, slip neglected	18,540 cubic feet.
Water pumped in 12 hours	1,388,100 gallons.
Pump horse power by pressure gauge.........	167·6.
Consumption of steam per hour	2,188 lb.
Steam per pump horse power hour............	13·05 lb.
Indicated horse power	183·70.
Steam per I.H.P. hour..........................	11·91 lb.
Mechanical efficiency of engine	0·913.

As will be seen, the steam consumption came out at much less than specified in contract.

Professor Unwin summarises as follows :—

"In this country the duty is estimated as the effective work of the engine in foot-pounds per 112 lb. of coal. The effective work in the trial was 167·6 × 1,980,000 = 331,850,000 foot-pounds per hour. The actual coal consumption was 296·5 lb. per hour. Hence the actual duty was 125,350,000 foot-pounds per 112 lb. of coal. This is a very good, but not exceptional duty."

"In this trial the efficiency of the boiler was not good, and the duty which depends on the performance of the boiler and engine is not so good as it would have been if steam had been supplied by a more efficient boiler. With a

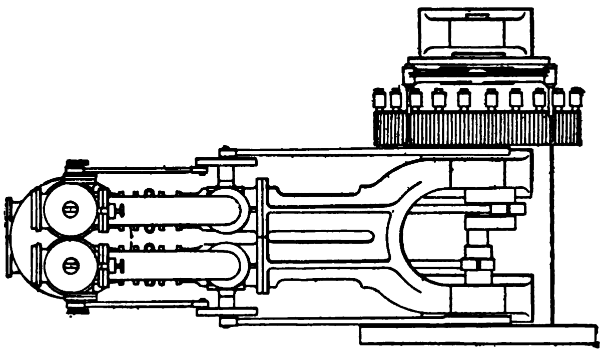


FIG. 88.

good boiler, hand fired, with Welsh coal, the evaporation might very well have been 9·5 lb. per pound of coal. Then the coal consumption would have been 240·8 lb. per hour. In that case the duty would have been 154,350,000 foot-pounds per 112 lb. of coal. This is an exceptionally high duty."

"In America it is common to reckon the duty of a pumping engine as the foot-pounds of effective work per 1,000 lb. of steam supplied to the engine. Taking this measure, the duty of the engine is 151,670,000 foot-pounds. This is almost as high a duty as has ever been recorded. It involves no assumption as to the performance of the boiler."

In the illustration, at fig. 87, the letters F.L. indicate the engine room floor level.

Riedler Pumping Engines.

In a former article, when discussing pump valves generally, we referred briefly to the Riedler mechanically-operated valves for the water end of a pump.

Figs. 88 and 89 represent in plan and elevation respectively a Riedler pump in direct connection with an electric motor. The general arrangement of the operating mechanism for the water valves is clearly indicated in the figures.

The illustrations are from the catalogue of Messrs. Fraser and Chalmers Limited, the sole manufacturers both in this

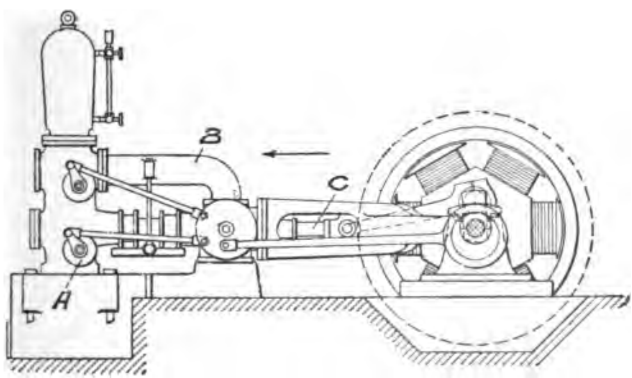


Fig. 89.

country and in America of the Riedler pumps. Their English works are at Erith, Kent, and their London office at 3, London Wall Buildings. The particulars given hereunder are from the maker's "description and argument" concerning the Riedler system:—

"The principal feature of the Riedler pump is its mechanically-operated valve. The valve and valve seat are circular in form, and made of high grade bronze. The

valve has a lift of from 1 in. to 2 in., and an area of such amount as to reduce the speed of the water flowing through same to but a few feet per second. At the beginning of the stroke the valve opens automatically, controlled, however, by a very simple and effective mechanical device. It remains open practically the entire stroke. When near the end it is positively closed at the proper moment by the controller. The valve opening being large, all throttling of the water through the valve passages is avoided. The mechanical controller, closing the valve at the proper moment, prevents slip, and allows the pump to be run at any desired piston speed."

"In ordinary pumps, owing to the large number of small valves required to get the desired valve opening, and the fact that these valves are closed by a spring in conjunction with the reversal of pressure, in case it is desired to run the engine at a high number of revolutions the lift of the valves would necessarily have to be reduced, or otherwise an enormous loss due to slip would result. Reducing the lift of these small valves would necessitate the use of a greater number. This would result in making the pump end much larger. As it is at present, the pump ends on the ordinary pumps are often built with large flat surfaces, the form least able to withstand water pressure. Or if the circular form of pump body is used, it would mean making this of much greater diameter, so as to be able to place in same a large number of so-called valve cages. In either case the strength of the pump body is reduced unless a large addition is allowed to the thickness of the same. This results, of course, in the ordinary direct-acting and fly-wheel pumps running at a low piston speed, otherwise the loss due to throttling of the water through the numerous valve passages, already great, is made greater, and loss by slip largely increased."

"In the Riedler system, however, because of the circular form of the valve, and the fact that all pressure parts are built cylindrical or spherical in form, it is simply necessary to slightly increase the diameter of the valve and its lift, thus greatly increasing the valve opening, and but slightly decreasing the strength of the pump body."

"It should be borne in mind that the difficulty of running pumping engines at high speed is due to two things, namely: the small valve area, necessitating the travel of water through it at a high velocity; and high lift, for unless valve is closed mechanically a great loss by slip will result. These difficulties are entirely overcome by the construction of the Riedler valve and its mechanically-controlled operations."

A record of a test made in the year 1895 on a pair of double-acting Riedler pumps arranged at the rear of, and driven directly by, a cross-compound condensing Corliss engine, gives a duty of 125,824,903 foot-pounds per 1,000 lb. of dry steam. The high-pressure cylinder of engine is 22 in. and the low-pressure cylinder 36 in. diameter; the pump plungers are 15¾ in. diameter, and the common stroke 42 in. Steam pressure, 120 lb. by gauge. The capacity at 75 revolutions per minute is 15,000,000 gallons (U.S.) per 24 hours (representing a mean piston speed of 525 ft. per minute) against a pressure of 62 lb. per square inch.

At a test made by Professor Edward F. Miller, of the Massachusetts Institution of Technolgy, in May, 1895, on a Riedler pumping engine at Chestnut Hill Pumping Station, Boston, U.S.A., the extremely low figure of 11·22 lb. of steam per indicated horse power per hour was recorded. The mechanical efficiency is given at 89·46 per cent, so that the steam consumption per actual or pump horse power comes out at 12·54 lb. per hour, or a duty of 158,147,000 foot-pounds per 1,000 lb. of steam supplied to the engine. The following description is given concerning the above pumping engine:—

"The engine proper is a vertical inverted beam engine of the Leavitt type, having cylinders 13·7, 24·3, and 39 in. in diameter, and a stroke of 6 ft. Steam is admitted to the high-pressure cylinder at 185 lb. pressure (200 lb. absolute), directly from a separator which forms the inlet side pipe. Re-heaters are placed between the high-pressure and intermediate, and betweeen the intermediate and low-pressure cylinders, and are supplied with steam at boiler pressure. The high-pressure and intermediate cylinder and cylinder heads are steam jacketed with the steam at boiler

pressure, and the low-pressure cylinder and cylinder heads are steam jacketed with the steam at 100 lb. pressure. The drains from the high-pressure and intermediate jackets, and from the re-heaters, are led directly back to the boiler, while the drains from the low-pressure jackets discharge into the feed-water heater. From the low-pressure cylinder the exhaust passes through a surface condenser, which is served with water from the discharge of the main pumps."

"The pump end is of the Riedler type, with mechanically-operated valves, and consists of three double-acting plunger pumps, each with its two inlet valves and two discharge valves. At the side of each pump there is a wrist plate, through which the valves are controlled by power derived from the engine. By this mechanism the spring pressure is removed from the valves previous to their opening, and is re-applied toward the end of the plunger stroke in such a manner as to bring the valves easily to their seats at a rate in proportion to the diminishing flow, thus preventing a back flow and the forcible closing of the valves with the reversal of the action. Suitable provision is made for elasticity in the connection to allow for any obstruction to the closure of the valve. The action of the valves is thus rendered so easy that the pump runs smoothly, and easily, at over 50 per cent above its rated capacity."

The average steam pressure in boiler during trial is given at 175·7 lb., and the speed of engine and pump 50·59 revolutions per minute, being a piston speed of 607 ft. per minute. The record of water pumped during 24 hours' trial is 21,016,000 U.S. gallons against 137 ft. head.

Comparing this Riedler triple-expansion vertical pumping engine with the foregoing vertical triple-expansion pumping engine by Messrs. Hathorn, Davey, and Co., it will be noted that, although the cylinder diameters closely correspond, the Riedler engine works with a higher steam pressure, and has double the length of stroke, and runs at a piston speed two and three-quarter times greater than that of the Hathorn-Davey engine. The tests show a 4 per cent advantage in steam consumption in favour of the Riedler engine, but we have no figures before us regarding the respective first costs of the engines, and the expense of maintaining the same in effective condition.

Referring again to the figs. 88 and 89, the pump there shown in direct connection with the motor is described as a "duplex differential" type. As will be seen, the two water cylinders are each divided, and an externally-packed ram is reciprocated between them, stuffing boxes being arranged on the adjacent ends of the divided parts. There is but one suction and one delivery valve for each complete pump cylinder, the said valves being arranged in the rear division. On the movement of the ram in the direction indicated by the arrow at fig. 89, the water drawn in through the suction valve A on the preceding out-stroke is displaced, and whilst one portion of it is forced direct into the rising main, another portion flows along the pipe B, and into the forward division of the cylinder, to fill the void space resulting from the difference in area between the outgoing portion of the ram and the incoming portion of the rod C. On the next out-stroke the suction water flows into the rear division of the cylinder, and a delivery is effected from the forward division into the rising main. Messrs. Fraser and Chalmers state: "The cost of a direct-connected electric motor of a stated power, by reason of its low speed, is greater than that of a motor of same power but of higher speed; by direct connecting, however, all gearing, belting, or rope transmission, with attendant evils, are avoided: thus the greater convenience, durability, and economy of the direct-connected electric motor in a very short time more than save the difference in the first cost."

They also give the following figures as to the actual losses in the electric transmission of power:—

"10 per cent loss due to friction of steam engine.

"3 per cent loss between engine and generator, due to belting. This loss does not appear if direct connected.

"10 per cent loss in electric generator.

"From 10 to 20 per cent loss in line depending upon voltage, length, size of conductors, kind of installation, and care of installation.

"12 to 20 per cent loss in the motors, depending upon the type and the service they are to perform.

"5 to 10 per cent loss between the brake horse power of the motors and the machinery that is to be driven.

"Loss in the machine itself."

This leaves a total efficiency of about 50 per cent. The results as obtained above are practical working results, for which we are indebted to several of the large mining companies that are using electricity, and who also have other machinery built by ourselves. No account has been taken in the above of losses due to step-up, step-down, and rotary transformers. Where distances are short, and great care is taken in the installation, the actual working efficiencies for electricity may be increased to 60 per cent, or, in case of carelessness in installation, may be decreased below 40 per cent."

Fig. 90 represents in outline one of four compound-condensing beam type rotative pumping engines, as constructed by Messrs. Gimson and Company, of Vulcan Street, Leicester, for the sewage pumping station of that borough. The engines afford an excellent example of what may be accomplished with a comparatively low steam pressure (80 lb.) by careful design and sound construction.

The particulars given hereunder are from a report presented by Mr. E. G. Mawbey, the surveyor of Leicester, to his Town Council in the year 1891.

"The sewage is delivered from the pumping station through two 33 in. rising mains for a distance of about a mile and a half into the distribution tanks at the farm, to a net height of about 163·66 ft. above the invert of the outfall sewer at the pump wells. There are four engines of the independent rotative compound-condensing beam type. The diameter of the high-pressure cylinder is 30 in., with a stroke of 5 ft. 9¼ in., and that of the low-pressure cylinder 48 in., with a stroke of 8 ft. 6 in. The cylinders are steam jacketed, that of the high-pressure cylinder being fed with steam at the boiler pressure of about 80 lb. to the square inch, and that of the low-pressure cylinder at the pressure at which it leaves the high-pressure cylinder. The steam cylinder slide valves are of the double-piston type, arranged in a cylindrical casing, cast separately from the cylinder, through which the admission and escaping steam for both high and low

pressure cylinders passes. The high-pressure steam cylinders are fitted with expansion piston valves, arranged to cut off the steam at any point from $\frac{5}{8}$ to $\frac{1}{8}$ of the stroke by means of hand gear, which can be worked whilst the engines are running, the working rate of the expansion being clearly

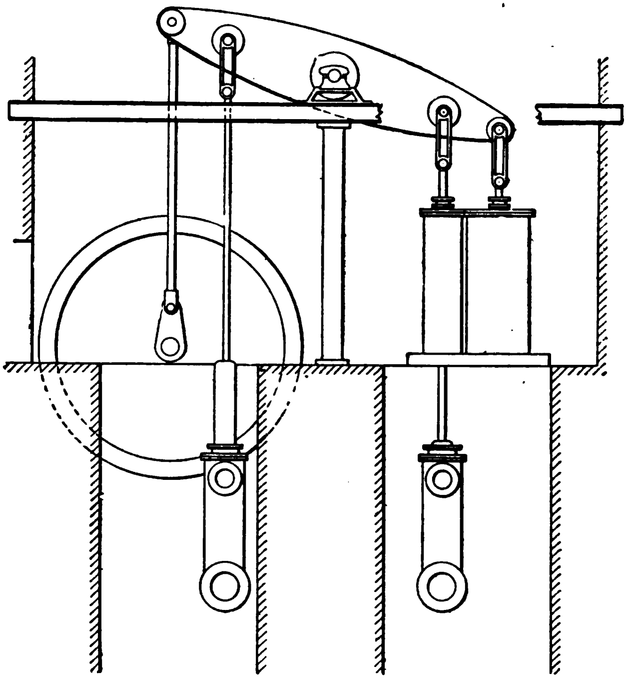


FIG. 90.

indicated by automatic arrangements of pointers and scales. The low-pressure cylinders are also fitted with expansion piston valves, arranged to allow of the cut-off being varied between $\frac{1}{4}$ and $\frac{1}{2}$, the working point being automatically indicated as in the case of high-pressure cylinders.

"To each engine there are two main pumps for the sewage of the piston and plunger type, one at each side of the beam, having a stroke of 5 ft. 9¼ in., the diameter of the piston being 27½ in. The suction and delivery valves are flaps, faced with indiarubber, and the hinges bushed with gun metal. The two main suction pipes are 3 ft. in diameter, leading from the pump well and screen chamber to each pair of engines. A large steel air vessel 25 ft. 9 in. by 5 ft. diameter is fixed to each rising main. The air pumps and condensers are of the single-acting jet type, the interna- valves and fittings being of gun metal, with flat indiarubber discs.

"The flywheels are of cast iron, 21 ft. in diameter; they weigh about 21 tons each. The beams are formed of double steel flitches, 2 in. in thickness and 6 ft. in depth at the centre."

As will be seen from the illustration, the pump rod for the pump on one side or end of the beam is formed by a continuation of the high-pressure piston rod.

The specification called for an effective duty of not less than 100,000,000 ft.-lb. per 112 lb. of coal, with the engine running at a speed of 12 revolutions per minute, and a boiler pressure of 80 lb. per square inch. The record of the official tests gives an average duty (measured by the weight of water actually pumped) as 115,913,333 ft.-lb. per 112 lb. of coal (Nixon's Navigation). As the average evaporation in the boilers per pound of coal (from actual temperature of feed water and at actual steam pressure) was 10·03 lb., the duty per 1,000 lb. of steam works out as follows:—

$$115{,}913{,}333 \times \frac{1000}{112 \times 10\cdot 03} = 103{,}184{,}493 \text{ ft.-lb.}$$

The average indicated horse power during the trials was 199·63, and the average pump or actual horse power 178·47; the mechanical efficiency was thus 89·41 per cent.

Measured in horse power terms, the average coal consumption was 1·712 lb. per indicated and 1·915 lb. per actual or pump horse power per hour. The steam consumption was 17·17 lb. per indicated and 19·2 lb. per pump horse power per hour.

It is interesting to note that the "slip" through the valves and the pistons of the pumps was but 1·27 per cent with one pair of the engines, and 0·87 per cent with the other pair.

By a separate test, the coal required for the boiler feed pumps was found to be 2·38 per cent of the whole quantity used, or 0·05 lb. per actual or pump horse power. The

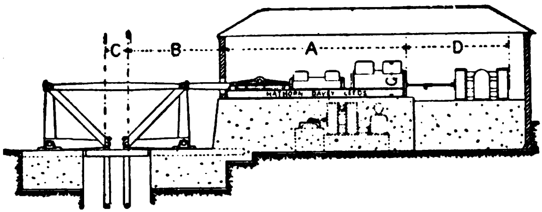


FIG. 91.

steam used for the feed pumps during the trial was taken from a separate boiler, supplied with coal apart from that used for the pumping engines.

MINE PUMPING ENGINE, SURFACE TYPE.

Figs. 91 and 92 represent, in elevation and plan respectively, a differential surface mine pumping engine by Messrs. Hathorn, Davey, and Company, of Leeds. Such

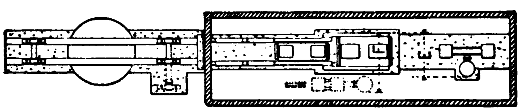


FIG. 92.

engines, which may be either single-cylinder, compound, or triple expansion, are fixed on the surface, and their reciprocating motion is transmitted to the pumps (fixed in the shaft) by means of quadrants mounted on beams across the shaft mouth. The particular type illustrated is termed by the makers the "South Staffordshire mine drainage type," and the engine is arranged in this instance for working

bucket pumps in pairs. For lifts up to 200 ft. an engine capable of raising 1,400 gallons of water per minute will occupy the following (approximate) space: A, 28 ft. 6 in.; B, 15 ft.; C, 3 ft. 9 in.; D, 16 ft.; E, 7 ft. 3 in.; F, 11 ft. 3 in.; G, 6 ft. 3 in.

The pump shown in the illustration is a compound with jet condenser, the cylinders and condensers being disposed in a tandem arrangement as illustrated.

The advantages claimed by Messrs. Hathorn, Davey, and Company for surface as against underground pumping engines are: Economy in steam, and therefore in coal; in boilers, and in stokers' wages; also greater safety in cases where the mine is liable to be flooded.

Mine Pumping Engine, Underground Type.

Fig. 93 illustrates, in elevation and plan, a differential underground mine pumping engine, by Messrs. Hathorn, Davey, and Company. The engine is compound and jet condensing; the two steam cylinders, the condenser, and the pumps (which in this case are of the piston type) are

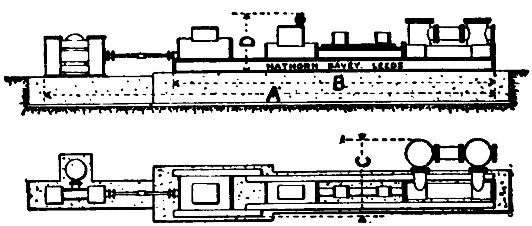


Fig. 93.

disposed in a tandem arrangement, as illustrated. The makers express their experience regarding piston pumps as follows: "Pumps of this class are less suitable for permanent colliery work than ram pumps, as, unless the water is practically free from grit, the piston packing is liable to give trouble; but, on the other hand, they occupy a rather smaller space longitudinally than a ram pump, and are somewhat less in first cost." For heads up to 150 ft. a

pumping engine of this type capable of raising 1,300 gallons per minute will occupy the following (approximate) space: A, 43 ft. 6 in.; B, 31 ft. 6 in.; C, 7 ft. 9 in.; D, 5 ft. 6 in.

The makers give as the advantages of underground pumping engines, as compared with the surface type, that their capital cost is much less, probably about one half the amount for the same horse power, and that space is not taken up in the shaft with spear rods and guides. They are, therefore, very largely adopted where coal is cheap and capital cost an important consideration.

As regards the duty of the two types, Messrs. Hathorn, Davey, and Co. make the following statement: "As an approximate estimate, the duty in pounds of water raised 1 ft. high per cwt. of ordinary engine coal burnt may, for single or duplex steam pumps, be taken as from 12 to 25 millions according to the size of pump, steam pressure, and conditions of work; and for a compound steam pump from 30 to 40 millions; while a surface engine will in ordinary work give a duty of from 50 to 60 millions."

Whilst such statements from eminent firms setting forth their experience of the figures obtained in the daily working of pumps are of great practical value, insistence must again be laid on the importance of having before us particulars as to the respective services and conditions of the rival types before arriving at any definite conclusion.

Davey's Differential Valve Gear.

The valve gear from which the Hathorn-Davey pumping engines, above referred to, obtain the name of "differential" is illustrated at fig. 94. The makers' description is as follows:—

"Davey's differential gear consists essentially of a small subsidiary engine, the speed of which can be regulated by means of a cataract cylinder to any desired rate, and of a pair of links having no fixed anchorage, but attached at one end to a rocking shaft (or other convenient part) driven from and moving with the main engine, and at the other to the small subsidiary engine above referred to. It is evident that, supposing the subsidiary engine and the main engine

to be moving in opposite directions, a point at the centre of the links will not be moved exactly in accordance with either of them, but will have a mean or differential motion between the two. It is from this central point that the valves receive their motion, and the arrangement is such that the valves are opened when the links are moved in the direction in which the subsidiary engine tends to move them and closed when the links are moved in the direction in which the main engine tends to move them. As before noted, the subsidiary engine is controlled by a cataract cylinder, and can, therefore, be set to move at a rate which

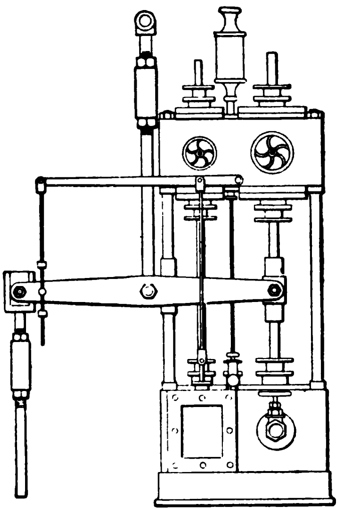


FIG. 94.

will give the necessary valve opening when the main engine is working at any required speed; but directly the speed is exceeded, either from increase of steam pressure or decrease of load, the main engine gains upon the subsidiary engine, the closing of the valves is accelerated, and steam either throttled or entirely cut off. The gear therefore forms a sensitive governor acting directly upon the steam valves. The illustration shows a subsidiary engine of the kind above

referred to, together with a second small subsidiary engine also regulated by a cataract. This second small engine is used to reverse the steam valve of the differential gear engine, which therefore pauses, and with it the main engine also, until the secondary engine has made its stroke. The cataract regulation enables this pause to be adjusted to a nicety, from a mere dwell at the end of each stroke to a pause of 10 to 15 or more seconds when for any reason it is desired to run the main engine dead slow."

THE WORTHINGTON HIGH-DUTY PUMPING ENGINE.

Fig. 95 is an illustration of one of four Worthington horizontal triple-expansion high-duty pumping engines installed for the entire water supply at the Paris Exhibition of 1900. The following particulars are from the published records of the Worthington Pump Company :—

"Each engine has—

Two high-pressure steam cylinders,			each	12 in.	in dia.
Two intermediate	,,	,,	,,	20 in.	,,
Two low-pressure	,,	,,	,,	34 in.	,,
Two double-acting water plungers			,,	26 in.	,,

And all have a uniform stroke of 24 in.
Size: 12 and 20 and 34 by 26 by 24."

The engine is capable of delivering 6,600 imperial gallons of water per minute against a head of 32 lb. per square inch with a steam pressure of 150 lb., and when working at a piston speed of about 150 ft. per minute.

The high-pressure cylinders are bolted directly to the cradle between the steam and water ends; the intermediate cylinders are placed next, but with an intervening space between. The low-pressure cylinders are bolted directly to the intermediate cylinders, and thus form the extremity of the steam end. The high-pressure piston rods are directly coupled to the pump rods. The low-pressure pistons are attached by side rods to their respective crossheads. Each intermediate piston is connected to its low-pressure piston by a rod working through a long metallic sleeve. The intermediate-pressure cylinders are fitted with dash relief valves to regulate the length of stroke of the engine.

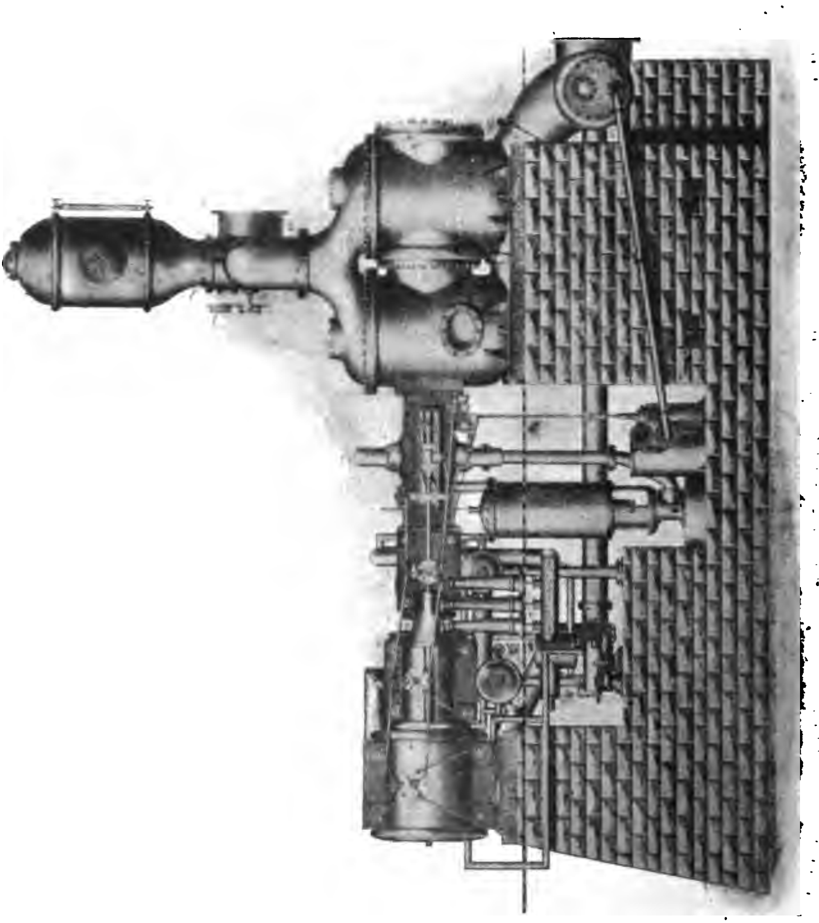


FIG. 95.

The steam in passing from one cylinder to the next goes through re-heaters situated below the high and intermediate cylinders, where it is re-heated by live steam at full boiler pressure. These re-heaters are fitted with brass tubes and arranged in connection with the jacket system, so that the steam passes in succession through the cylinder jackets and re-heaters.

The steam-valve motion is a modification of the Corliss type. The cylinders have two admission valves, one at each top corner, which serve also as cut-off valves; the two exhaust valves of each cylinder are placed at the bottom corners. The four valves are operated from a central wrist plate, moved through the medium of links and rockshaft from the opposite crosshead, in the ordinary manner of the duplex valve gear. The exhaust valves are opened and closed by links connecting the cranks on the valve spindles directly with the wrist plate. The steam admission and cut-off valves, on the other hand, are connected by links from their cranks to a secondary four-arm crank, fulcrumed on the wrist plate, but which receives its motion from its own side of the engine. This gives the effect of a broken-link or knuckle-joint connection between the admission valve and the wrist plate, which results in the valves being opened by the motion of the wrist plate proper as derived from the opposite side of the engine, and closed by the secondary motion carried through the four-arm crank and derived from their own side of the engine. The makers claim that with this arrangement the cut-off is as rapid as can be desired, and that being without trip motion, releasing, or other device, the valve gear is exceedingly simple, and capable of very wide adjustment. The point of cut-off may be varied at will, and each valve can be altered separately while the engine is in operation.

The four compensation cylinders, two for each side of the engine, are of cast steel, and have large trunnions carried in bearings on the main frames. Each cylinder contains a single-acting cast-iron ram or plunger, having chilled and ground surfaces. These plungers are screwed into T heads or thrust pins, which work in bearings carried on the respective crossheads of the main piston rods. Thus with the

motion of the piston rods the compensating cylinders are oscillated back and forth by means of their plungers, which run in and out of the stuffing boxes. The compensating cylinders are always under constant pressure; hence there is an equal load on the compensating plungers at all points of the stroke, and the actual effect of this force on the piston

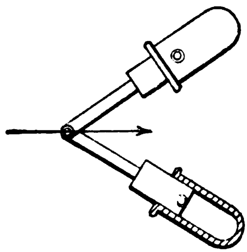


FIG. 96.

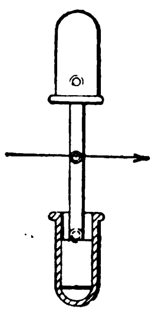


FIG. 97.

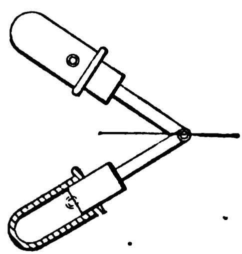


FIG. 98.

rod of the engine is determined by the various positions of the compensating cylinders during the stroke. The closed ends of the compensating cylinders are connected by ports which pass through the trunnions to a central distributing pipe which is in connection with a differential accumulator.

Figs. 96, 97, and 98 represent in diagram form the positions of the compensating cylinders at the beginning (fig. 96), the centre (fig. 97), and the end (fig. 98) of a

piston stroke. During the first half of each stroke of the engine pistons the compensating cylinders exert a retarding influence. At mid-stroke they have no effect on the motion of the pistons, but they assist the motion in the second half of stroke, and such accelerating influence increases uniformly with the decrease in the force exerted by the expanding steam.

Though the head or pressure on pumps was rather under 31 lb. per square inch, they gave a pump horse power per hour with 14·97 lb. of steam. The particulars supplied concerning the test giving such result are as follow :—

Steam pressure	156 lb.
Superheat	91·8 deg. Fah.
Water pressure	30·5 lb.
Vacuum	26 in.
Piston speed	162·4 ft. per minute.
Pump horse power	156
Steam consumption (exclusive of jackets).................	13·35 lbs. per P.H.P. per hour.
Steam consumption (inclusive of jackets).................	14·97 lbs. per P.H.P. per hour.

The makers give the following general particulars concerning their compensating system :—

"By thus alternately taking up and exerting power due to the different angle in which their force is applied to the line of motion of the plungers, these compensating cylinders in effect perform the function of a flywheel, but with the important economical difference that they utilise a pressure of compressed air instead of the energy of the momentum. Their action is readily controlled, and their power may not only be proportioned to the work to be done, but it is entirely unaffected by the speed of the engine; and the same amount of expansion may be obtained by the engine whether running at a speed of 10 or 200 ft. per minute, or whether running at 5 or 50 revolutions. This latter feature is one of great importance, affecting as it does so favourably the economy of the engine when applied to any service where the demand is irregular or intermittent. Where such service is performed by a flywheel engine it is a well-known

fact that the best economical results are obtained when the engine is running at its full rated capacity, and that this economy rapidly diminishes as the speed is decreased. With every change made in the speed of a flywheel engine a corresponding change must be made in the point of cut-off; when the speed is decreased the steam must be made to follow further in the stroke of the piston, thus reducing the expansion, and consequently the efficiency and economy of the engine. There are many flywheel engines running on direct or standpipe systems that are obliged to go slowly in delivering the quantity of water demanded by the service, so as to necessitate the steam being carried nearly, if not quite, full stroke. As a result their practical duty falls off very materially, oftentimes not exceeding one-half the duty obtained by trial.

"The Worthington high-duty engine with compensating cylinders overcomes this objection perfectly. The economy is not appreciably affected by the variations in its speed, and as the rate of expansion of the steam in cylinders is constant under all changes in the rate of delivery or speed of the pump, the economy can only be affected as regards the greater cylinder condensation due to the slower speeds, and which, in an engine as thoroughly jacketed as this type of engine always is, can only be a very small percentage. The force of the compensating cylinders can, at the will of the attendant, be thrown on or off the engine instantly, and without the cut-off mechanism becoming disarranged; they can be quickly disconnected from the engine, which can then be run as economically and as satisfactorily as those ordinarily constructed without them."

CHAPTER XIV.

Hydraulic Rams.

These well-known machines, with which the energy of a body of flowing water can be utilised for the purpose of elevating a portion of such body to a higher level, are eminently suited and largely adopted for the water supply of large country residences, and for farms, nurseries, or other like services where a river or stream is available. Hydraulic rams are also constructed, so that what we may term the motive-power water, instead of raising a portion of itself as aforesaid, is made to raise well water to the required elevation. This type is sometimes known as the "dirty-water ram."

Fig. 99 is a sectional elevation representing one form of the ordinary hydraulic ram, in which the motive-power water elevates a portion of itself. Such water, taken from a river or other source, flows down an inclined fall pipe (on opening the gate or stop valve in such pipe), enters the ram at A, and passes out through the inwardly opening valve B. But the rush of water speedily closes the valve B, and the outlet being thus blocked, the momentum of the body of water within the inclined fall pipe and the ram itself lifts the valve C, thus permitting the water to pass into the air vessel D, and up the rising main or pipe E. When the momentum of the water is spent, the valve B again falls open, and the action is repeated. This will continue as long as the fall-pipe valve is open and water is available for keeping such pipe charged. The pulsations, or openings and closings of the valve B, follow in regular and rapid succession, so that with a good air vessel a fairly regular delivery may be obtained through the discharge pipe E. F represents a small snifting valve, admitting a little air at each pulsation.

In the "dirty-water ram," represented in sectional elevation at fig. 100, the momentum of the water on the closing of the inwardly opening valve B is expended in moving the

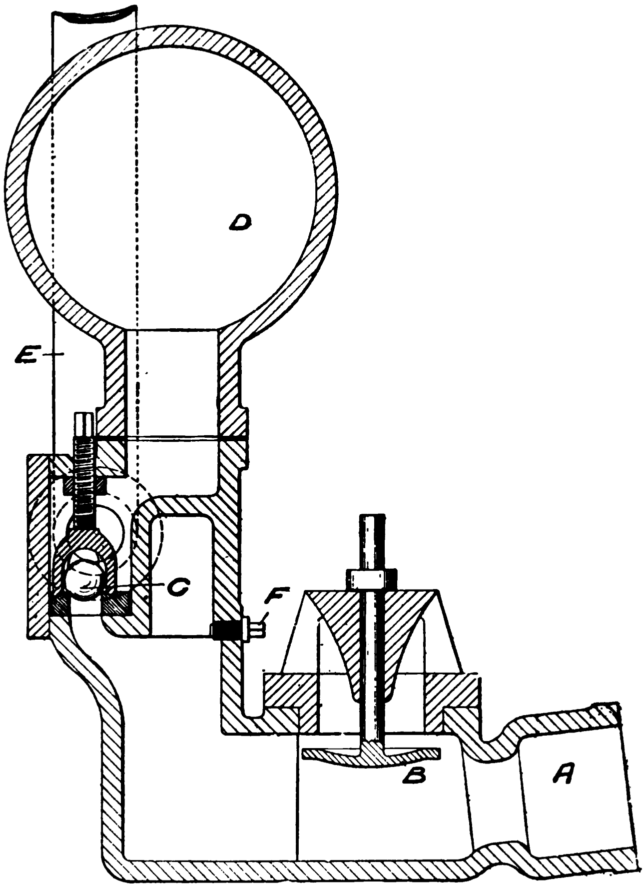


FIG. 99.

piston *a* against the action of the spring *b*. When the momentum is spent, the piston is returned by the spring,

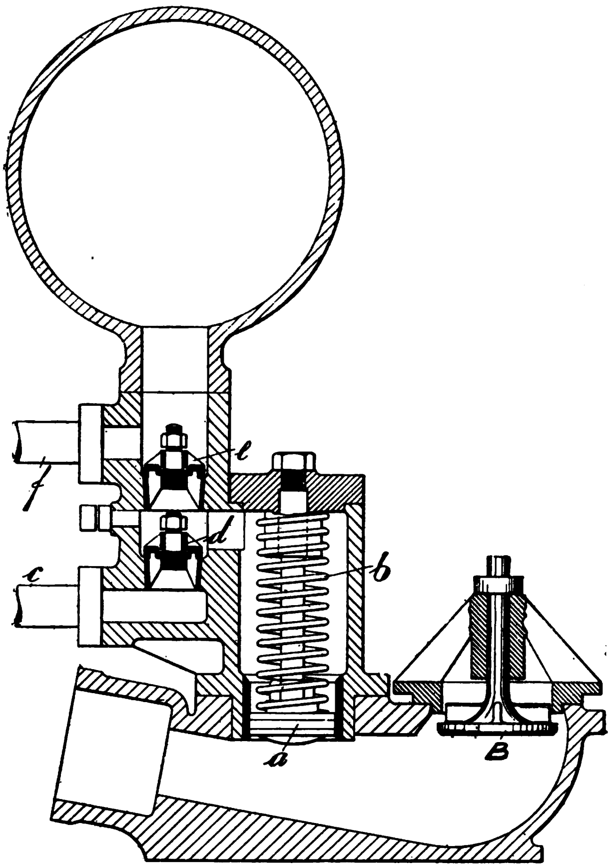


Fig. 100.

and thus we get, during the action of the ram, a continuous reciprocation of the piston *a*, which, like the ordinary dis-

placement of a pump piston or plunger, causes water to enter from the suction pipe *c*, through the valve *d*, and to be discharged through the valve *e* and pipe *f*.

Fig. 101 is an illustration of a hydraulic ram by Messrs. R. Warner and Co., of 97, Queen Victoria Street, London, E.C., and Walton-on-Naze, Essex. The makers give the following description:—

"They may be placed at any distance from where the water is required to be delivered, but care should be taken that for long lengths of rising main the latter should be sufficiently large, so that the power is not unduly wasted in overcoming the friction of the water through same. The quantities of water raised by hydraulic rams change very much, according to the ratio of the fall to the height that the water has to be raised. The drive pipes, to be satisfactory beyond the 2 in. size, should be of cast iron, made very heavy with face flanges, provided with strong brackets between the bolts, and the ends of the pipes should be turned and bored to fit each other for about three-eighths of an inch at the ends. Each ram is provided with an efficient snifting valve for maintaining a constant supply of air to the air vessel, a small quantity being admitted at each beat of the pulse valve."

Fig. 102 is an illustration of a ram termed the "Caliban," made by Messrs. W. H. Bailey and Co. Limited, of Albion Works, Salford, Manchester. The makers' list of sizes range from a $\frac{3}{4}$ in. fall or driving pipe and a $\frac{1}{2}$ in. delivery pipe, to a 6 in. fall pipe and a $2\frac{1}{2}$ in. delivery pipe. The approximate delivery to a height of 50 ft. with a 10 ft. fall is given as 25 gallons per hour for the small ram, and 1,500 gallons per hour for the large ram. The makers give the following further particulars:—

"The efficiency of all hydraulic rams depends chiefly upon the amount of fall to be obtained. They may be employed for such a slight fall as 2 ft.—that is, the spring, brook, stream, or other source may be only 2 ft. higher than the ram when fixed into position. As the height of the fall increases, however, the more powerfully does the hydraulic ram act, and its ability to force water to a greater elevation and distance increases correspondingly with the relative height

FIG. 101

of the source of supply above the ram; and the elevation to which it is required to raise the water, determines the relative proportions of the water raised and run to waste. For general requirements it may be safely calculated that

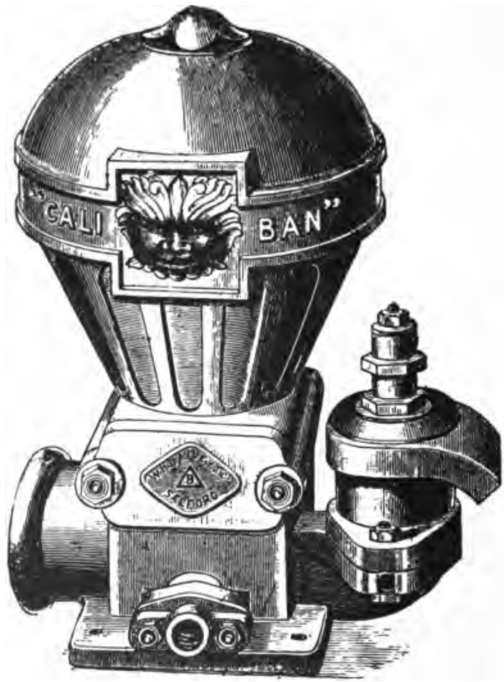


Fig. 102.

about one-seventh of the volume of water falling into the ram can be raised to an elevation five times the height of the fall, or one-fourteenth part of the volume can be raised about ten times the height it falls, and so on in like proportion according as the fall or height is increased or diminished.

"Bends or angles in the pipes (either fall or delivery) should be avoided if possible, and when necessary they should be as large as possible (bends preferable to angles). The pipes (both fall and delivery) should be run underground to protect them from injury. In cases where water has to be forced long distances, the delivery pipe should be rather larger than for short distances, to allow for friction, &c., of water in the pipe. To obtain the most effective duty from Bailey's 'Caliban' ram, the fall pipe should be placed at an angle of 45 deg. from the source of supply to the ram. When this is more than 12 ft. above, and the fall pipe descends at a greater angle than that of 45 deg., one or

FIG. 103.

more bends should be placed in the fall pipe, so as to reduce the velocity of the water, and thus prevent injury to the ram."

Another type of hydraulic ram by Messrs. Bailey is known as Decœur's patent. Fig. 103 is a small illustration giving an external view of one of these rams. Fig. 104 is a larger sectional view of one type of this ram, as described in the patent specification (No. 15731, of 1894). The specification states that waste is avoided by reducing the velocity of the current of water passing through the escape valve A and by placing the delivery valves B as near as possible to the escape valve.

The stroke of the valve A is controlled by the nut C regulating the tension of the springs c, c', and the stroke of the valves B by the nut D. The lever E serves to raise the valve A, and so start the ram. The waste outlet F is provided with a knee or bend (not shown) entering a water sump or basin which prevents re-entrance of air. Sufficient

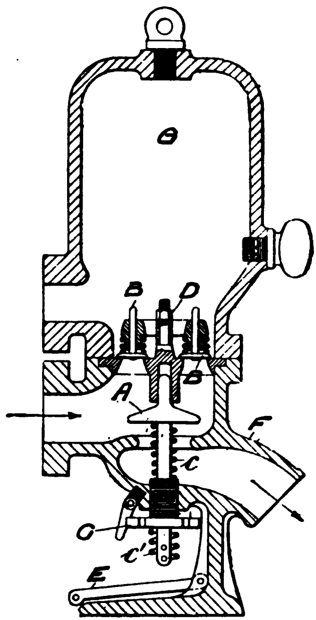


Fig. 104.

air for the supply of the air vessel or compression chamber G can pass around the valve rod. In large rams the valve A is replaced by a disc or ring-shaped valve of large diameter; the regulating springs are then placed above the chamber G, and connected to the valve by a sliding frame. Messrs. Bailey's list sizes of this type of pump ram range from a $\frac{3}{4}$ in. fall or drive pipe and $\frac{1}{2}$ in. delivery pipe, to a 20 in. fall pipe

and an 8 in. delivery pipe. With the latter size the capacity is given as 2,400 gallons per hour raised 100 ft. high with a 10 ft. fall; the mean quantity of drive water required to do this work is given at 600 gallons per minute. Referring to the advantages of this ram the makers state—

"It will force one-third of the drive water to two and a half times the height of fall, one-sixth to five times the fall, one-tenth to eight times the fall. It is suitable for any fall from 12 in. to 40 ft."

It is frequently, but erroneously, concluded that hydraulic rams can never at any time utilise more than a small percentage of the energy of the fall or driving water. In the trade lists published by Messrs. Blake and Sons, of Accrington, they record the following result of a test by a user:—

"Working fall of driving water, 30 ft.; vertical height raised, 127 ft.; length of rising main, 850 ft. from ram to outflow; length of supply pipe, 200 ft.; gallons per hour raised, 1,612; driving water used per hour, 8,186 gallons. Efficiency, 83 per cent."

The efficiency has been obtained by comparing the useful work done with the actual expenditure of energy required to do it. Thus it is evident that as the 8,186 gallons, or 81,860 lb., of driving water fell 30 ft., its energy on entering the ram was

$$81860 \times 30 = 2455800 \text{ foot-pounds};$$

similarly, the useful work done was

$$16120 \times 127 = 2047240 \text{ foot-pounds};$$

we thus obtain the efficiency as follows:—

$$\frac{2047240}{2455800} = \cdot 83.$$

Another published result as to the performance of a ram of a different make gives an efficiency of but 27·63 per cent, with a fall of 11 ft. and a rise of 76 ft.

From the figures given concerning the last sized Decœur ram previously referred to, it will be seen that the energy of one hour's flow of fall water is

$$600 \times 10 \times 60 \times 10 = 3,600,000 \text{ foot-pounds};$$

and the actual work done

$$2400 \times 10 \times 100 = 2{,}400{,}000 \text{ foot-pounds};$$

these figures give an efficiency of

$$\frac{2400000}{3600000} = \cdot 67.$$

In comparing figures relating to the performance of rams, it must be carefully remembered that the higher the ratio between the fall and the height of lift, the lower the efficiency. The figures published on a list of ordinary type rams by another maker, for a 10 ft. fall and a 100 ft. lift, gives an efficiency of but 50 per cent, as against the above-named 67 per cent for the Decœur ram; but if the same ram is employed for a 50 ft. lift with the same fall, the efficiency rises from 50 to 79 per cent.

Windmill Pumps.

The application of windmills for the purpose of raising water, so largely practised in America, is steadily, if slowly, extending in this country. The annular sail type is that most generally used. Fig. 105 represents such a mill, 10 ft. diameter and 20 ft. high, as made by Messrs. Robert Warner and Co., of 97, Queen Victoria Street, London, and Walton-on-Naze, Essex. It is further described as "direct-acting," for no gearing is interposed between the sail shaft and the pump. The mill is provided with an automatic regulating arrangement for opening and closing the sails to allow the wind to pass through the same, in case of sudden gusts, without doing any damage. It can be stopped and started at any time. With a 10 mile wind the mill, as illustrated, and of dimensions above referred to, will raise 120 gallons of water per hour 100 ft. high. A mill having a sail 20 ft. diameter will raise 960 gallons under the same conditions.

Respecting wind power, Messrs. Warner give the following particulars :—

"We would remark that the quantities of water given in the lists, that our mills will raise, are less than those given by some other firms, but special attention should be paid

Fig. 105.

to the fact that the quantities we give are raised by a 10 mile wind, viz., just a pleasant breeze. This velocity of wind can generally be relied upon for 8 hours out of 24, while higher velocities cannot. It is highly important, especially where the storage capacity is limited, that the mills should, if possible, work every day. With a 20 mile wind, viz., a brisk breeze, windmills are four times more powerful than with a 10 mile wind, and our mills with about a 15 mile wind blowing through the entire 24 hours have often raised from four to five times the quantities we give for a wind blowing at the rate of 10 miles per hour, reckoning upon the latter being available for 8 hours out of the 24 only. The large quantities, however, that can be raised in strong winds are, in the majority of cases, of little or no value, owing to insufficient storage capacity obtainable."

A wind velocity of 10 miles per hour, or 14·67 ft. per second, gives a pressure of nearly $\frac{1}{2}$ lb. per square foot; 15 miles per hour gives a pressure of rather more than 1 lb. per square foot. The ratio between the pressures may be taken as about 2 : 5.

The list diameters of the sails of direct-acting windmills by Messrs. Warner and Co. range from 7 ft. to 20 ft. The geared windmills have sails ranging from 16 ft. to 40 ft. in diameter. With the largest size, 6,000 gallons of water per hour can be raised against 100 ft. head with a 10 mile wind.

CHAPTER XV.

Manual and Animal Power Pumps.

James Watt, for the purpose of comparing his engines with animal power, adopted a rate of work equal to 33,000 foot-pounds per minute as his standard measure of the capacity of a horse. Though this remains the recognised standard in this country for the expression of mechanical work, it must not be forgotten that Watt wisely adopted a high value. Any animal worthy of the name of a horse should be capable on emergencies, and with the judicious application

by the driver of the recognised persuasive and coercive methods, to perform work, for a very short time, much in excess of the engineers' standard. But only a most powerful cart horse will be able to keep up such standard rate of doing work for a lengthened period, as may be necessary on pumping service.

Similarly, though a strong man working at "high pressure" may be able for a short time, and by a special effort, to exert $\frac{1}{6}$th of a horse power, or to do mechanical work at the rate of 5,500 foot-pounds per minute, the standard "man power" as adopted by leading makers of hand pumps, viz., $\frac{1}{12}$th H.P., or 2,750 foot-pounds per minute, is doubtless high enough for any continued effort.

And great care must be taken not to assume too high a figure in considering the force or pressure exerted by a man on a pump handle. It is possible for him to impose the entire weight of his body upon the handle, but the force or pressure exerted by an ordinary man on continuous pumping should not be estimated at more than 25 lb. If the leverage of the handle is insufficient to give the required increase of such pressure (of course at the expense of the stroke), then gearing must be provided between a hand wheel and the pump rod to still further increase the pressure applied to the bucket or plunger, with a corresponding reduction in speed.

A hand wheel, or flywheel as it is sometimes termed, is generally considered more pleasant to operate than an ordinary pump handle, as the former, especially if provided with a heavy rim, serves as an equaliser of the energy applied thereto.

The following figures are from the catalogue of Messrs. Joseph Evans and Sons, of Wolverhampton :—

Capacity of	a powerful horse	=	33,000	ft.-lb.	per min.
,,	,, pony or mule	=	16,500	,,	,,
,,	,, an ox	=	11,000	,,	,,
,,	,, a man	=	2,750	,,	,,

Fig. 106 is a typical hand pump for general outdoor service. The water is not forced through piping or subjected to pressure by the upstroke of the bucket, but

is simply lifted by such stroke up to the level of the spout through which it is discharged. Such a pump is therefore known as a "lift pump." The makers of the pump, as illustrated—Messrs. Joseph Evans and Sons, of Wolver-

FIG. 106.

FIG. 107.

FIG. 108.

hampton—also describe it as a "yard pump." As an anti-freezing arrangement the makers place the working barrel, 3 ft. long, under the ground level, instead of making the hollow standard itself serve as the barrel, as in the figure.

Fig. 107 represents Messrs. Evans' "flywheel lift pump with compensating lever," which the makers state "will be

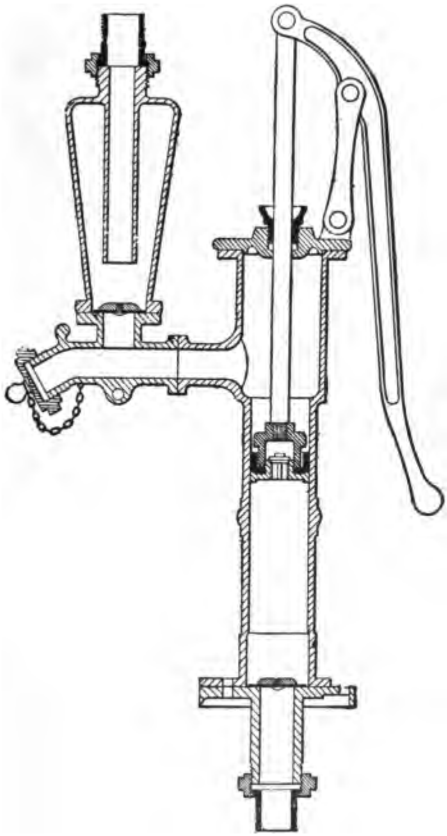


Fig. 109.

found to work much easier than the ordinary crank-motion pump."

In the "lift and force pump," by Messrs Evans, illustrated at fig. 108, the pump rod passes out of the delivery chamber through a stuffing box, and the said chamber is so closed as

FIG. 110.

FIG. 111.

to permit of the application of such force upon the water therein as to discharge it through the outlet branch under the required pressure. Fig. 109 is a sectional view of a pump of this type, with the delivery branch provided with a

Fig. 112.

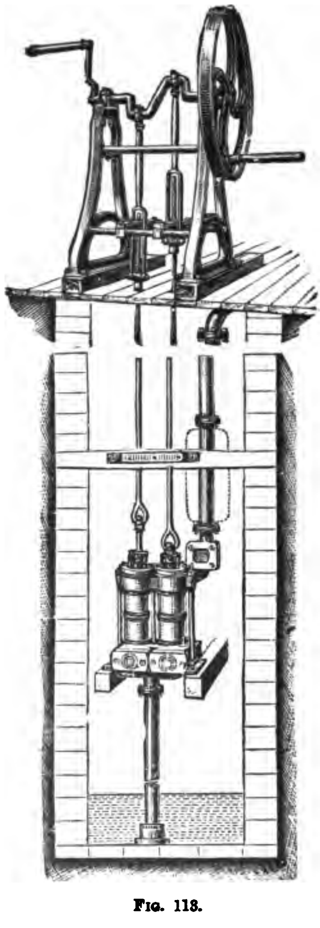


FIG. 118.

screwed end for hose connection, and delivery pipe connection with attached air vessel and dip pipe.

Fig. 110 illustrates a lift and force pump, by Messrs. Evans, mounted on a hard wood plank. This well-known type of hand pump can be very compactly and conveniently fixed against a wall.

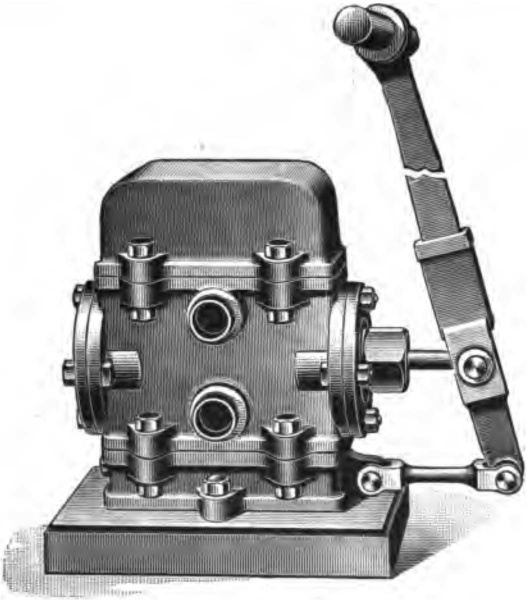


FIG. 114.

The lift pump illustrated at fig. 111 is by Messrs. Robert Warner and Co., of 97, Queen Victoria Street, London, and Walton-on-Naze, Essex. They term it their "Pillar-frame" type. Fig. 112 illustrates combined manual and animal power well-pumping machinery by the same makers. Either side can be put in or out of gear as required by means of lever-operated clutches.

Fig. 113 illustrates a double-barrel deep-well pump by Messrs. Evans, with what is termed an engine frame, carrying

the operating crank shaft provided with handles and flywheel.

Another type of hand pump is shown at fig. 114. This is known as the "Manchester" double-acting hand pump, by Messrs. Frank Pearn and Co. Limited, of West Gorton, in that City.

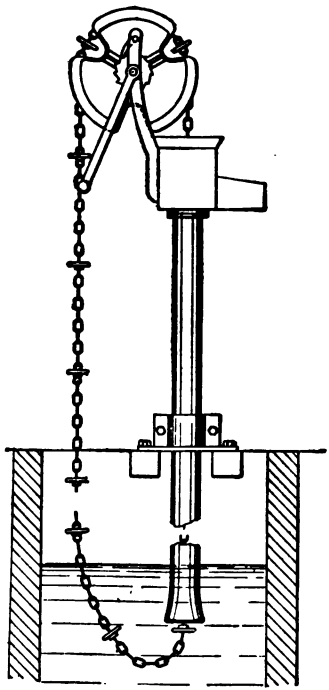


FIG. 115.

Fig. 115 represents a modern type of one of the most ancient of water-raising appliances—the chain pump. The particular machine here shown is by Messrs. Robert Warner and Co. Having no valves or packing of any kind, these extremely simple machines are employed with

advantage for pumping liquid manure, gas tar, thick muddy water, and for many other services that would speedily clog

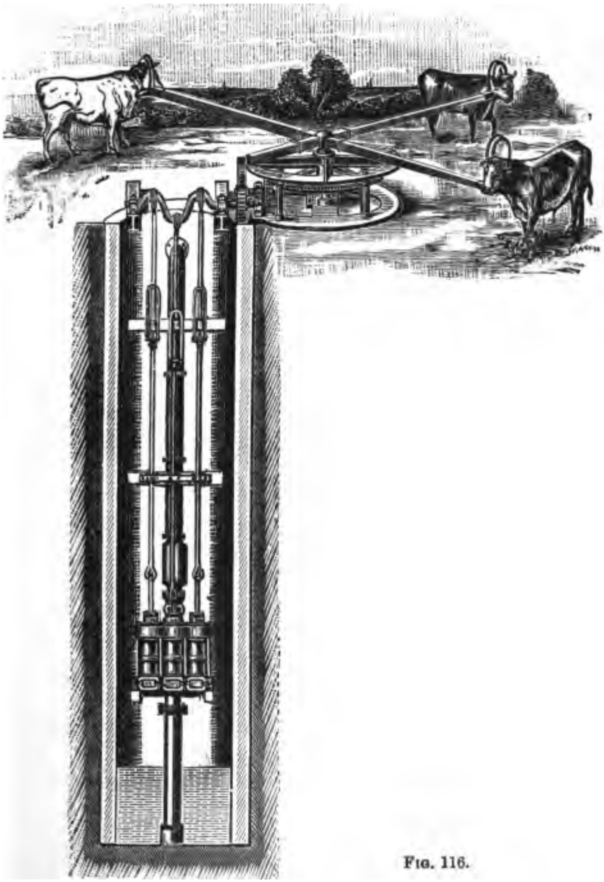


FIG. 116.

an ordinary reciprocating pump. Messrs. Warner also make chain pumps arranged for animal or for engine power driving.

Fig. 116 is an illustration of treble-barrel deep-well pumps, with bullock or horse gear, by Messrs. Joseph Evans and Sons.

A centrifugal pump arranged with horse gear, by Messrs. W. H. Allen, Son, and Co., of Bedford, is illustrated at fig. 117. The makers recommend it for supplying country houses with water, for filling cattle tanks, and for similar purposes. They state: "It is very easily fixed and soon

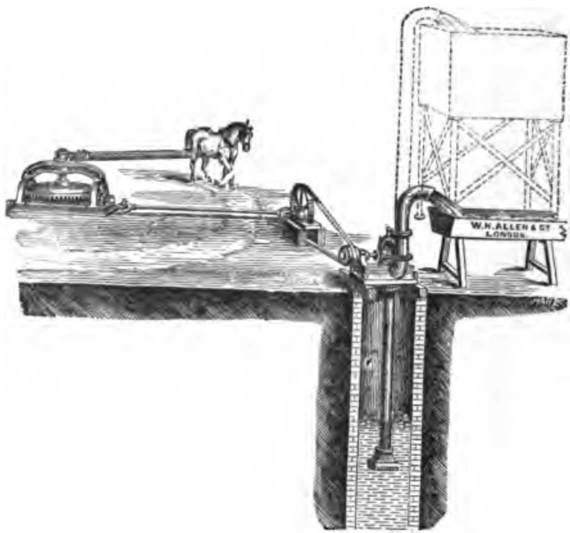


FIG. 117.

set to work, it is not likely to get out of order, and any ordinary labourer can attend to it. The horse gear can be used for driving chaff-cutting or turnip-cutting machines, and other tools when not pumping."

SEMI-ROTARY PUMPS

A very neat and compact type of double-acting hand-power pump is obtained by the employment of fixed and movable radial brackets or valve plates within a circular casing. The

brackets are each provided with a pair of valves, serving for the suction and delivery respectively. The movable bracket, which carries the suction valves, has a rocking motion imparted to it by means of a lever fitted on the projecting end of the central spindle to which the bracket is secured. The said rocking motion gives the displacing or pump action. The pump can be readily bolted to a wall or plank.

Diaphragm Pumps.

In these pumps, which are used for dealing with thick and muddy liquids—indeed, "with anything that will flow," as is sometimes stated—the displacing action is obtained by the reciprocation of the central portion of a circular or other shaped diaphragm. Such diaphragm consists of a flexible disc of leather, indiarubber, or other material, having its edges fixed to the pump body. The space beneath or on the one side of the diaphragm forms the suction chamber, whilst the opposite side serves as the delivery chamber. A valve and its seating are fitted to the central portion of the diaphragm, and also a bridle or bridge piece, to which the pump rod is attached. The reciprocation of the pump rod with the attached central portion of the diaphragm is effected by a pump handle or lever in the ordinary manner.

PART II.

CHAPTER XVI.

HIGH-DUTY PUMPING ENGINES—PRESENT PRACTICE.

THE present standard practice, both in this country and in America, for high-duty pumping engines for waterworks and similar services, is to employ triple-expansion engines of the vertical and three-crank type, the pump rams or plungers being worked direct from the engine crossheads.

An excellent example of the foregoing is the pumping engine at the Rosario Waterworks, Argentina, built by Messrs. Hathorn, Davey, and Co. Ltd., of Leeds. The steam cylinders are respectively 15 in., 25 in., and 40 in. diameter, and the stroke of each piston 36 in. In direct connection with each crosshead there are two single-acting rams, one of 16 in., and the other of 16¾ in. diameter. The rams or plungers have, of course, a uniform stroke of 36 in., the same as that of the steam pistons. The normal speed of the engine is about 35 revolutions of the flywheel shaft per minute, giving a piston speed of 210 ft. per minute. The makers give the results of a test as follows: Horse power in water lifted, 209; boiler pressure, 177 lb. (above atmosphere); steam used per I.H.P. per hour, 11·09 lb.; thermal efficiency measured in indicated work, 20·99 per cent; steam used per pump H.P. per hour, 12·28 lb.; combined mechanical efficiency of engines and pumps, 90·2 per cent; duty per 1,000 lb. steam, 161,250,000 foot-pounds. The steam was superheated to the extent of 53·3 deg. Fah. at the engine stop valve. From the foregoing figures the thermal efficiency in actual work done can be calculated as follows:—

$$20{\cdot}99 \times \frac{90{\cdot}2}{100} = 18{\cdot}93 \text{ per cent.}$$

A further interesting example is afforded by the very much larger engine, of the same type as the foregoing, at the

Boston (U.S.A.) Waterworks, and constructed by the Allis-Chalmers Company, of Milwaukee. The cylinders are respectively 30 in., 56 in., and 87 in. diameter; stroke, 5½ ft.; capacity, 30,000,000 U.S. gallons per 24 hours; with 185 lb. steam pressure, and 195 ft. piston speed. The following figures are given as the result of an official test: Dry steam per I.H.P. per hour = 10·335 lb.; duty, 178,497,000 foot-pounds per 1,000 lb. of dry steam (this gives 11·09 lb. per pump horse power per hour, and a mechanical efficiency of 93·1 per cent); thermal efficiency, measured in pump horse power or actual work done, 21·63 per cent.

With regard to the foregoing, it will be seen that the advantage in steam consumption per pump horse power of the Boston over the Rosario engine is 9·7 per cent. It will probably be agreed, however, that the makers who could produce such an excellent result with a comparatively small engine, such as that of Rosario, would have no trouble in obtaining that 9·7 per cent with an engine giving the possibility of economy presented by the figures representing the size and capacity of that at Boston. But no well-advised waterworks committee in this country is likely to adopt so large an engine. Economy, both in first cost and in maintenance, will, after due allowance for the advantage of the large engine in steam consumption, point in the direction of two or more smaller engines rather than the abnormally large one.

Fig. 118 is from a photograph of a triple-expansion pumping engine by Messrs. Hathorn, Davey, and Co. Ltd.; the engine is similar to that at Rosario, above referred to. Fig. 119 illustrates an engine of the same type, by the same makers, but in this case the flywheels are placed outside instead of between the cranks; the engine is one of three supplied to the Birmingham Canal Navigation.

Messrs. Hathorn, Davey, and Co. Ltd. are at the present time building four triple-expansion vertical pumping engines, which they have guaranteed to work with a steam consumption of 11½ lb. per pump horse power per hour The makers may be relied upon to obtain a duty appreciably under their guarantee.

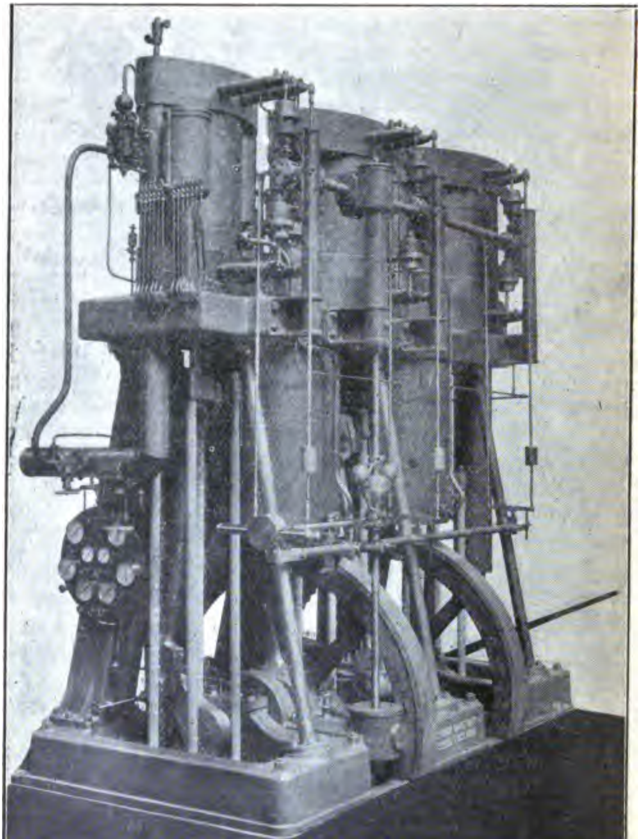


FIG. 118.

In the engines previously referred to the pumps are below the crank shaft and the floor level of the engine room. Messrs. Glenfield and Kennedy Limited, of Kilmarnock, have recently erected at a pumping station on the Wilge River, South Africa, two sets of triple-expansion vertical pumping engines, in which the pumps are above the crank shaft, being fixed on the engine back columns, below the slide guides. Each crank is driven by a connecting rod, which is forked; the pump barrel lies between the two arms of the fork. Each engine has steam cylinders 16½ in., 27 in., and 45 in. diameter, with a 2 ft. stroke. The single-acting rams are each 10⅜ in. diameter. Each engine is fitted with an Edwards air pump, worked from the crosshead of the intermediate cylinder. The surface condensers are in pits in front of the engines; the water from the suction main passes through the condenser before reaching the pumps. The steam pressure is 160 lb. There is one flywheel for each engine fitted at one end of the crank shaft, which does not overhang but is supported by an out bearing.

In comparison with the foregoing vertical type triple-expansion pumping engines provided with crank shaft and flywheel, we may refer to the twenty sets of horizontal type non-flywheel engines (Worthington) for the Coolgardie (Western Australia) water supply, which are briefly described as follows: Twenty sets of triple-expansion high-duty Worthington pumping engines, each having two high-pressure cylinders 16 in. diameter, two intermediate cylinders 25 in. diameter, and two low-pressure cylinders 46 in. diameter, all having a common stroke of 36 in. These engines were required for pumping water over a distance of about 350 miles. Distributed in eight pumping stations along the pipe line, twelve of the sets have water plungers 21 in. diameter; the remaining eight have 15 in. plungers. The contract was secured by Messrs. James Simpson and Co. Ltd., of Grosvenor Road, London. Working under superheated steam at a pressure of 175 lb. per square inch and a temperature of 500 deg. Fah., the contract provided that each of the pumping engines should be capable of attaining, throughout a 12

hours' trial, a duty of 135,000,000 foot-pounds of effective work per 1,000,000 B.T.U. supplied to the engine, which could not be returned to the boiler in the ordinary course of working. In published records of the tests the duty per 1,000,000 B.T.U. figures at 143,000,000 foot-pounds, whilst the work obtained from each 160 lb. of coal (having a calorific value of 10,000 B.T.U.) reached 144,427,000 foot-pounds in one test and 148,141,000 foot-pounds in another test. Taking the mechanical equivalent of the British thermal unit as 772 foot-pounds, the thermal efficiency measured in actual work accomplished by pumping is as follows:—

$$\frac{143{,}000{,}000}{772 \times 1{,}000{,}000} = 18{\cdot}52 \text{ per cent.}$$

The steam consumption in terms of the horse power is not given by the published figures. It would appear to be about 12½ lb. per pump horse power per hour.

The Relative Thermal Efficiencies of Steam, Gas, and Oil Engines.

In connection with the foregoing figures as to triple-expansion steam engines for pumping service, it is interesting to compare them with the results obtainable from gas and oil engines. The figures already cited for the triple-expansion steam pumping engine at Boston show a thermal efficiency in actual work done of 21·63 per cent. The mechanical efficiency is 93·1 per cent, and the indicated thermal efficiency will therefore be

$$21{\cdot}63 \times \frac{100}{93{\cdot}1} = 23{\cdot}23 \text{ per cent.}$$

The indicated thermal efficiency of a large high-class gas engine may be as much as 35 per cent. For the Diesel oil engine an indicated thermal efficiency of 41 per cent has been reported. But the cost of pumping cannot, of course, be calculated merely from indicated thermal efficiencies. With a triple-expansion steam pumping

engine, having the steam pistons and water plungers in direct connection the one with the other, a very high mechanical efficiency is obtainable. Mr. Michael Longridge, in a discussion on a paper on the Diesel oil engine

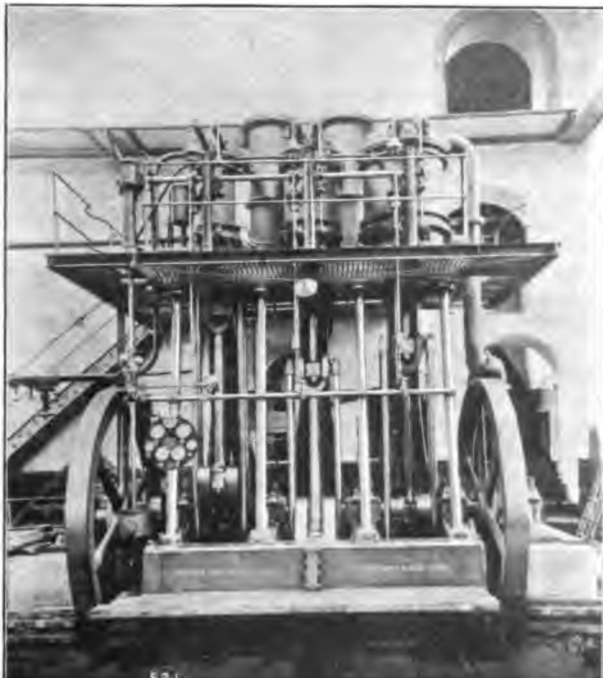


FIG. 119.

(Proceedings, Institution of Mechanical Engineers, No. 3 of the year 1903), referred to trials he had made of a Diesel engine rated at 30 B.H.P., in which the indicated thermal efficiency ranged from 23·8 per cent to 39·1 per

cent. But the mechanical efficiency with the last-named value of the indicated thermal efficiency was only 62·2 per cent. Thus, in that case the thermal efficiency, measured in actual work done or in brake horse power, is—

$$39 \cdot 1 \times \frac{62 \cdot 2}{100} = 24 \cdot 3 \text{ per cent.}$$

It will be seen, therefore, that a very high indicated thermal efficiency may not always represent the saving that the pump user may anticipate.

To obtain the high indicated thermal efficiency values to which we have previously referred in connection with gas and oil engines, it is necessary to employ a piston speed much higher than is suitable for the water plungers of a pump. Hence combined gas engines and pumps have not hitherto been constructed with the pump plungers in direct connection with the piston rod crossheads, as with the steam pumping engines already referred to. But pumps of the turbine type, to which reference will be subsequently made, appear to be as suitable for direct connection to a gas-engine crank shaft as to the armature shaft of an electric motor. For some services such a pump, directly driven by a gas engine, would probably prove more economical, even for pumping large volumes of water against considerable heads, than any high-duty steam pumping engine as at present constructed.

Details Relating to Reciprocating Steam Pumps or Pumping Engines.

Superheating, Re-heating, Steam-jacketing.—For many years the well-appreciated advantages of superheating were considered to be outweighed by the attendent disadvantages, arising chiefly from burnt packings and difficulties in the way of lubricating the internal moving parts of the engine, owing to decomposition of the oil. But to obtain the previously cited thermal efficiencies with steam pumping engines super-

heating must be resorted to, and as suitable packings and oils are now readily obtainable, it is commercially advantageous to superheat to a moderate degree, so that

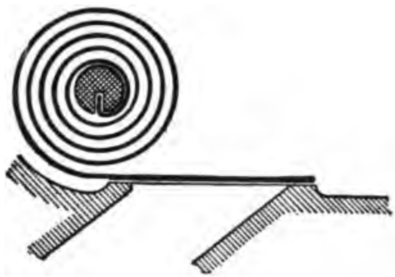


FIG. 120.

the steam shall be kept sufficiently above its saturation point to avoid much if not all cylinder condensation.

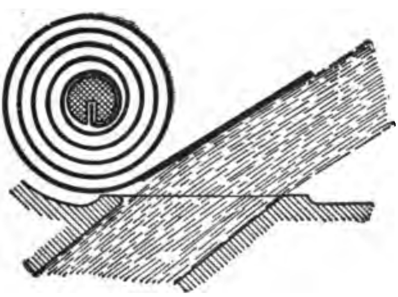


FIG. 121.

At present it is not usual to superheat much beyond 100 deg. or 120 deg. Fah., and, as will have been seen from one of the examples we have previously cited,

excellent results can be obtained with an extent of superheating but little exceeding 50 deg. Fah. In tests and experiments engines of various types may have been worked with steam which by superheating has been brought up to or even exceeding a temperature of 1,000 deg. Fah. Engines working under such conditions have been not inaptly described as "steam gas engines," seeing that the steam is entirely a gas during the time of its passage through the engine, no condensation occurring and the exhaust steam itself being at a temperature above saturation or condensing point. With steam so

FIG. 122.

FIG. 123.

superheated there could scarcely be any need either for re-heaters between the respective cylinders of an engine or steam jackets on any of them. With saturated or dry saturated steam, or with but moderate superheating, re-heaters might be expected to be beneficial, but they do not always prove so. Steam jackets are normally supplied with steam at boiler pressure, but better results have been claimed (though no supporting evidence is available) by using boiler steam in the high-pressure cylinder jacket, high-pressure cylinder exhaust steam in the intermediate cylinder jacket, and intermediate cylinder exhaust steam in the low-pressure jacket.

With a view to increasing the thermal efficiency figures, and also for the benefit of the boilers, a **Feed-water Heater** is sometimes disposed between the low-pressure cylinder and the condenser.

Steam Valves.—In the construction of high duty pumping engines of the vertical rotative or crank and flywheel triple-expansion type, the English practice is to employ Corliss steam valves on all three cylinders. A similar practice is followed by the Worthington Companies in the construction of their duplex non-flywheel pumping engines. But with the first-named type of engine the general American practice is to employ Corliss valves on the high-pressure and intermediate cylinders, and poppet valves (of the Cornish equilibrium type) on the low-pressure cylinder; in some cases, however, the intermediate cylinder has Corliss admission and poppet exhaust valves. The German and other continental engineers avoid rubbing surfaces as far as possible on the high-pressure cylinder, particularly with superheated steam, and hence they prefer poppet valves for the high-pressure cylinder. In a paper on "American Pumping Engines" (Proceedings, Institution of Mechanical Engineers, No. 1 of 1905), Mr. John Barr suggested that as the low-pressure cylinders of American pumping engines were frequently of large diameter (90 in. or 94 in.), very large ports were required. Some of such cylinders have two poppet valves in the top and two in the bottom, giving large areas and small clearances.

Water Valves or Pump Valves.—There has been no general departure from the now long-established practice of using multiple valves, each valve consisting of a disc of metal, or of rubber or vulcanite backed by a metal plate, with a closing spring of brass or phosphor-bronze. In a large pump a very great number of such valves are employed. In the paper previously referred to, Mr. Barr instances a vertical triple-expansion rotative pumping engine, having three pump rams, each $33\frac{7}{8}$ in. diameter, in which the total suction and delivery valves amount to 1,176; each of the six valve casings contain

seven valve plates or cages, each cage being fitted with 28 valves. The suction and delivery valves are, in ordinary practice, identical.

Mechanically-operated or mechanically-controlled pump valves do not appear to have found the general favour amongst users that was at one time anticipated by some makers. The well-known Riedler mechanically-controlled valves permit of the high rotative speeds adopted in what are described as "express pumps." Such machines are well suited for power driving and for direct connection to an electric motor; but they have met with a formidable competitor in the "turbine pump," to which we shall subsequently refer.

The illustrations, figs. 120 to 125 inclusive, are from blocks kindly lent by Messrs. Fraser and Chalmers Ltd., of Erith, Kent, and 3, London Wall Buildings, London, E.C., relating to the "Gutermuth" patent valve. Referring to its construction, the makers state: "The valve can be made from a sheet of metal of the same thickness throughout, but in large valves subjected to heavy pressures the end of the sheet which forms the valve is thickened by a special process. The sheets are cut slightly wider at the end which is to be coiled, as this prevents the coils and the flaps from fouling the clamp, sides of the casings, or the adjacent valves. The valve is simply slipped on to a spindle, which is held in place by means of clamps or set screws. Allowing for the difference of purpose for which they are made, the construction of the 'Gutermuth' valve is very similar to that of the hair spring of a watch; both are equally elastic and pliable. As in the case of a hair spring, the 'Gutermuth' valve cannot be overstrained by any length of legitimate use. It may be claimed that this valve is similar to the hinge valve. Such is not the case, as its motion is altogether different, and at the same time there is no friction of any kind. Further, no separate springs or loose parts are necessary, as is the case with the hinged valve."

Fig. 120 represents the closed and fig. 121 the open position of the valve. "It will be seen,"

again using the makers' description, "that the valve moves out of the way of the moving fluid,

FIG. 124.

thus causing very small frictional losses and absence of wear and tear of the valve and its seat. The

valve is always placed at an angle with the port opening, by which means the port becomes entirely uncovered with a very small movement of the valve itself. The resistance of the valve to be overcome by the flow is very small, and, further, the peculiar lift of the valve causes it to close instantly when the flow ceases. The valve closes with a wiping motion, which distinguishes it from all other valves of this type. It is this motion which ensures the silent working of 'Gutermuth' valves."

Fig. 122 represents an ordinary spring-loaded mushroom or disc valve, and figs. 123 and 124 the same valve

Fig. 125.

when water is passing it under a slight pressure and a considerable pressure respectively. Fig. 125 is an illustration showing the valve removed, and the water issuing freely from the valve way. Messrs. Fraser and Chalmers state that from these illustrations (reproductions of instantaneous photographs) it will be seen how the flow of water is deflected by this ordinary type valve, causing considerable loss and wear and tear to the valve and seat.

The makers add that in all cases where the "Gutermuth" valve has been installed it has met with the greatest success, and that a large number of machines with other types of valves have been altered so that the "Gutermuth" can be employed, and the machines thus adapted to run at a great speed with high efficiency.

CHAPTER XVII.

Types of Reciprocating Pumps for Various Services.

Boiler Feed Pumps.—Fig. 126 illustrates the Weir tandem-compound feed pump, and fig. 127 a set of Weir twin-compound feed pumps. Relying on the known economy in steam consumption of their standard single feed pump, the makers, Messrs. G. and J. Weir Limited, of Cathcart, Glasgow, state that they do not ordinarily make the tandem-compound type for pumps of less than 6 in. diameter by 15 in. stroke, delivering 2,430 gallons per hour. Each pump shown is double-acting, having gun-metal water pistons with ebonite packing rings, and water valves of the makers' patent group type, the valves being of bronze and the seats of Admiralty gun metal. The twin-compound pump consists of two Weir pumps arranged, as shown at fig. 127, for compound working, but both pumps are also capable of working independently. The makers state that—"The arrangement has been found specially advantageous in electric power stations where there is a varying load. In the day time the high-pressure pump can be run singly; at night, when more power is required, the low-pressure pump can be connected, and the economy due to the compounding secured when the highest power is called for."

Figs. 128 and 129 illustrate a compound feed pump by Messrs. J. P. Hall and Sons Limited, of Peterborough. The makers supply the following results obtained from a test of such a pump having $6\frac{1}{4}$ in. and $9\frac{1}{2}$ in. steam cylinders, a $6\frac{1}{4}$ in. double-acting water piston, and a stroke of 15 in.

Boiler pressure.	Pressure on pump.	Total No. of double strokes.	Duration of test.	Weight of steam used.	Weight of water delivered.	Pounds of water delivered per pound of steam used.
			Mins.	Lbs.	Lbs.	
160	160	146	10	40	4,613	115·3
160	160	103	10	33	3,254	98·6
150	160	81	10	30	2,559	85·3

The first test is at full speed, viz., 14·6 double strokes per minute.

Second test is at three-quarter speed, viz., 10·3 double strokes per minute.

Third test is at half speed, viz., 8·1 double strokes per minute.

Taking 1 lb. pressure as equal to a head of 2·3 ft., the steam consumption per horse power per hour in the first test figures out as follows;

$$\frac{160 \times 2{\cdot}3 \times 461{\cdot}3}{33000} = 5$$

steam consumption per horse power per hour,

$$\frac{60 \times 4{\cdot}0}{5} = 48 \text{ lb.}$$

Ashley Pumps.—Fig. 130 is a sectional view, representing an Ashley pump, of one type, as made by Messrs. Glenfield and Kennedy Limited, of Kilmarnock, who give the following particulars: "The Ashley pump was designed to supersede the old form of bucket and bottom valve pump, and to remove, in so doing, two of the most serious difficulties experienced in connection with underground pumping. It is almost invariably the case in this class of pumping, notably in wells and boreholes, that the pump itself has to be placed at a great depth below the surface of the ground, and connected to the engine driving it by a corresponding length of pump rod. Moreover, in almost all cases when a stoppage occurs the water

FIG. 126.

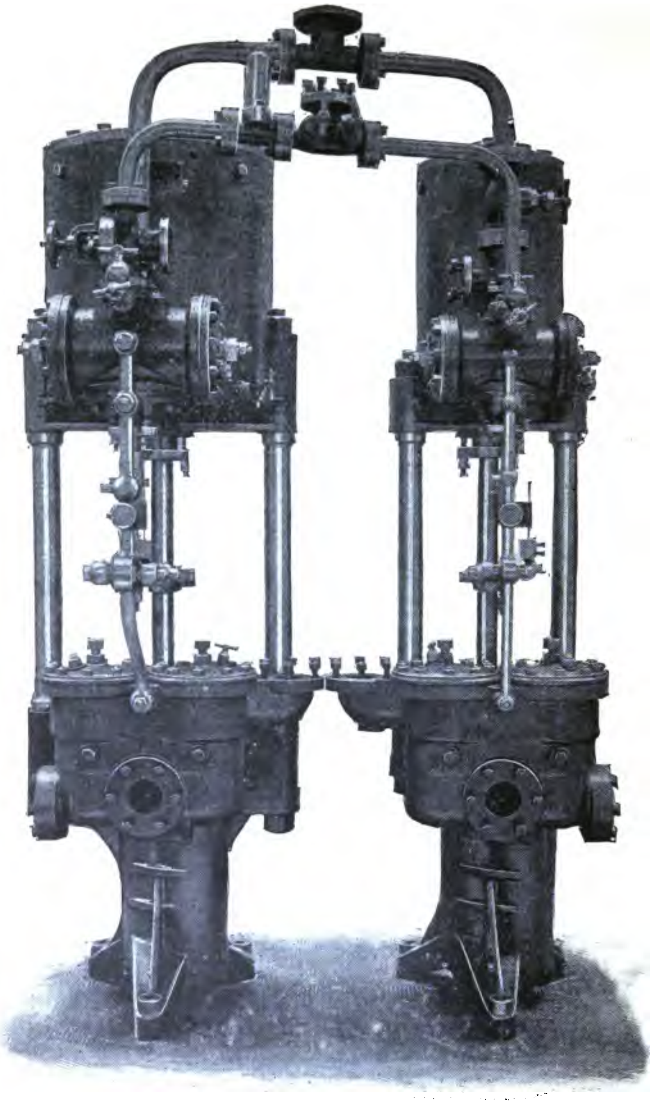


Fig. 127.

rises in the pumping shaft to a very considerable height (sometimes 200 ft. or 300 ft.) above the level at which the pump is fixed, and renders it quite impossible of approach from the outside for examination or repair. These conditions necessitate the use of a pump so constructed that all its working parts can be drawn up through the rising main when it is desired to effect any such examination or repair, and the diameter of the bucket must, therefore, be a little less than the diameter of the rising main through which it has to be drawn to the surface of the ground. This limitation, and the remote and confined position of the pump, have hitherto militated against the production of a pump for underground work giving anything like such good results and so little trouble as the best types of those used for surface work. For these latter as much room as may be desired can be obtained, and the working parts can be made accessible, and of good proportions, with very little trouble. The two great difficulties hitherto inseparable from the conditions of the problem, but which, we contend, are entirely removed by the 'Ashley' pump, are: Want of accessibility to the working parts, and slow speed. The principal advantages, therefore, that we claim are: 1. Perfect accessibility to, and improved arrangement of, working parts, resulting in freedom from breakdown and expensive repairs. 2. High speed (50 to 100 per cent faster than that of the ordinary bucket and bottom valve pump), quiet running, and great durability. The novel parts of the Ashley single-acting pump are two only, viz.: (1) The working barrel (W B, fig. 130). (2) The bucket (B, fig. 130), in which are contained all the working parts; not only the delivery valves, but also the suction valves."

The particular example illustrated by fig. 130 is what the makers term the shrouded type, used in those extremely rare cases—principally in sinking operations—where it is desired to work on suction.

Messrs. Glenfield and Kennedy Limited make the Ashley pump in many types to meet varying services,

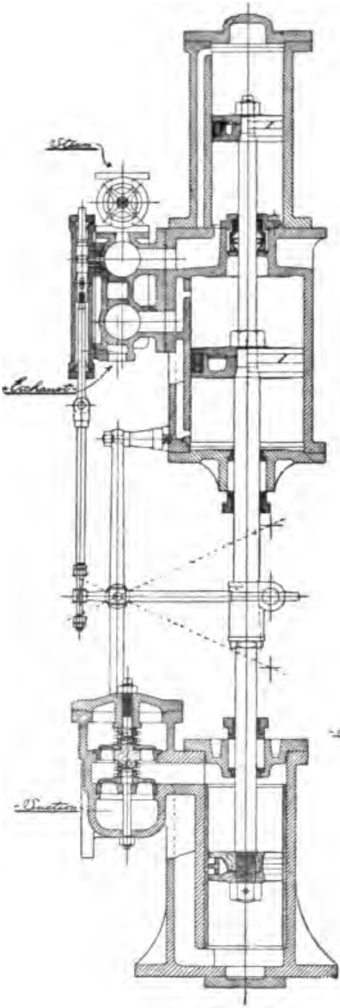


FIG. 128.

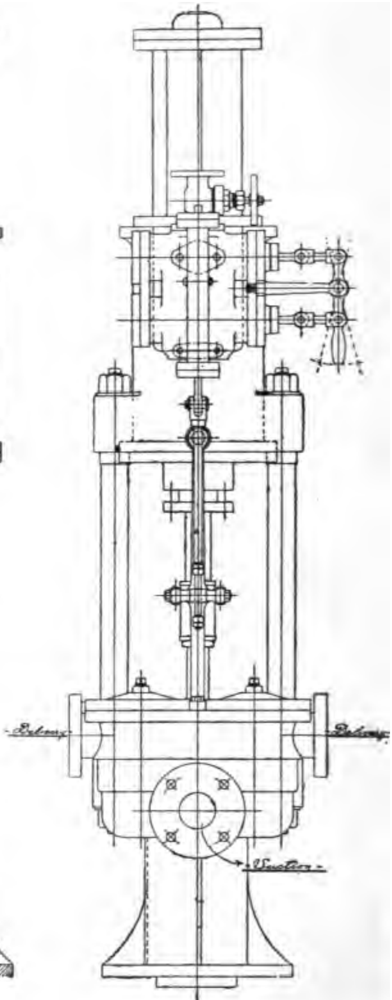


FIG. 129.

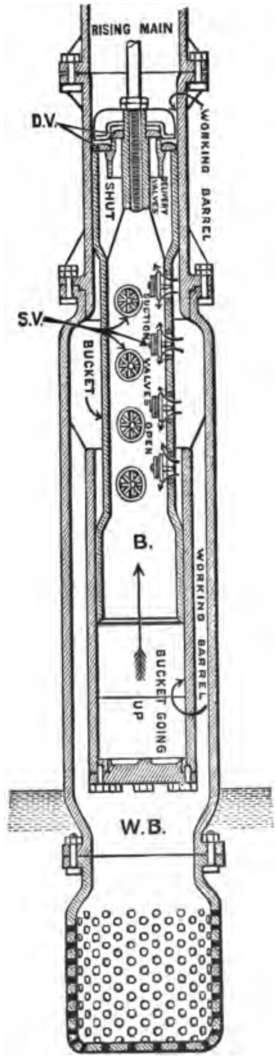

RISING MAIN
WORKING BARREL
D.V.
SHUT
DELIVERY VALVES
S.V.
SUCTION VALVES OPEN
BUCKET
B.
WORKING BARREL
BUCKET GOING UP
W.B.

FIG. 130.

and either single or double acting. The diameters of the pumps so far constructed range from $3\frac{3}{4}$ in. to 23 in., and the strokes from 1 ft. $5\frac{1}{2}$ in. to 4 ft. 6 in. A large number have been supplied to various waterworks throughout the country.

The "Hatfield" High-speed Pump.—Fig. 131 is an illustration of a high-speed reciprocating pump as made by Messrs. Merryweather and Sons Limited, of Long Acre, London, and known as the "Hatfield" pump. This pump is well adapted for direct driving by a high-speed motor. The example illustrated at fig. 131 depicts a combined pump and electric motor mounted on the one bed plate. The pump may also be driven by a petrol motor.

The "Hatfield" pump has three barrels arranged symmetrically around a central crank shaft; each barrel has a trunk-type water piston or plunger directly connected to the crank shaft. The said pistons are of comparatively large diameter and of short stroke; the pumps may be run at a speed as high as 700 revolutions of the crank shaft per minute. The valves are of indiarubber, and of the circular flap form, such as are employed by Messrs. Merryweather for their fire engines. The three pump barrels constitute part of a single triangular casting in which are also formed the valve chambers. To facilitate the starting of the motor a by-pass valve is fitted between the suction and delivery passages; on the opening of such valve only a very small load can be imposed on the motor. The pump illustrated was supplied to a country estate for the domestic water service and for fire protection. The pump and its motor form a very compact combination, occupying but little space, and being also silent in working, through the absence of transmission gears, it can, where necessary, be placed in a basement without causing annoyance to the occupants of the building.

Messrs. Merryweather arrange the "Hatfield" pump in various combinations to suit fire and other services. For fire boats the pumps driven by petrol or other motors may serve not only for the delivery of the required water

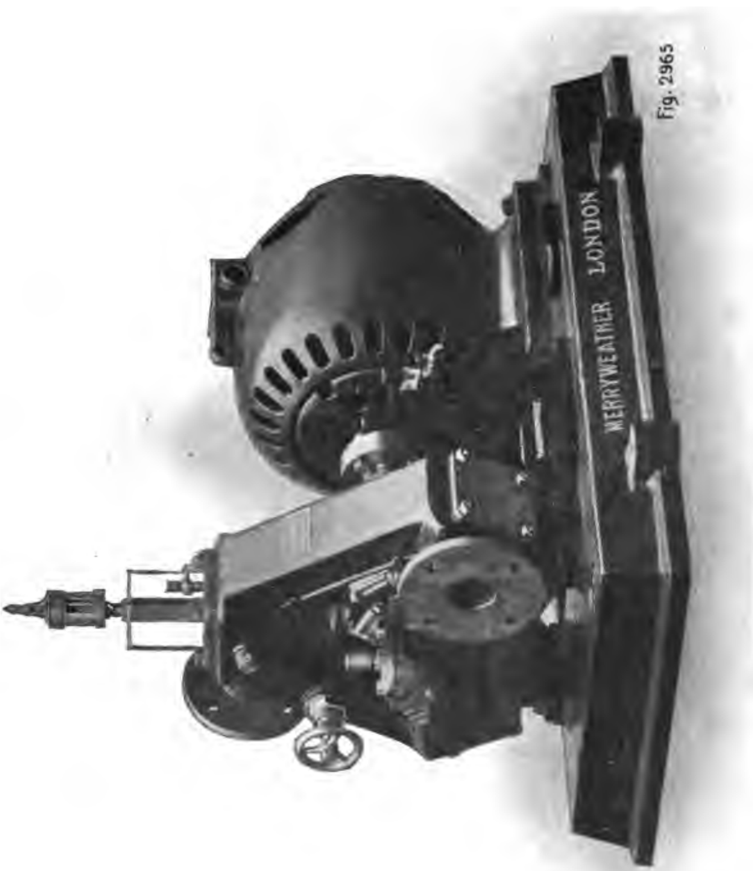


FIG. 131.

jets for fire extinguishing purposes, but also as the boat-propelling machinery. This is an example of hydraulic or "jet propulsion," for the boat is propelled by forcing water through four or other number of nozzles placed fore and aft at the sides of the boat.

To obtain varying capacities at constant speed the makers fit the "Hatfield" pump with a patent variable-stroke mechanism, comprising a sliding eccentric pro-

FIG. 132.—Diaphragm Pump by Messrs. Honig and Mock Ltd.

vided with internal gearing for adjusting the eccentric relatively to the crank shaft to obtain the required stroke alteration. For boiler feed and other services the gear can be arranged to permit of stroke adjustment while the pump is running. When working at a shorter stroke the pump is capable of giving a higher delivery.

Diaphragm Pumps.—Figs. 132 and 133 are illustrations of diaphragm pumps by Messrs. Honig and Mock Limited, of 7, Mark Lane, London, E.C. In these pumps, which are well adapted for dealing with thick, gritty, or muddy water, or for "semi-fluids" generally, the usual reciprocating piston or plunger is replaced by a diaphragm described by the makers as being "made of pure Para rubber," and which "will generally last for years." In the examples illustrated the reciprocation or breathing action of the diaphragm is effected by means of a hand

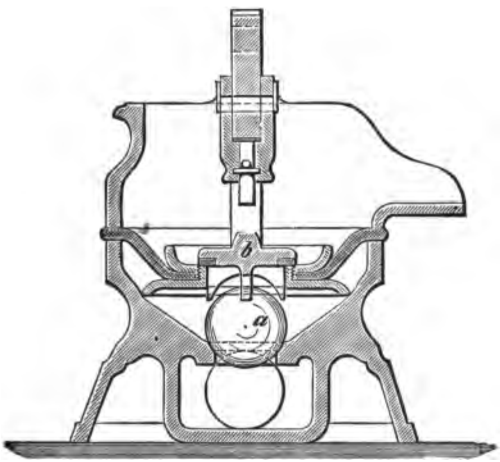


FIG. 133.—Detail of Valve of Honig and Mock Diaphragm Pump.

lever, but diaphragm pumps may also be driven by power. The suction valve *a* (fig. 133) is a rubber ball; the delivery valve *b*, of cast iron, has its under side recessed as shown to receive, and so prevent displacement of, the suction valve when the latter is raised from its seat. It will be seen that the passage from the suction to the delivery is as short as possible.

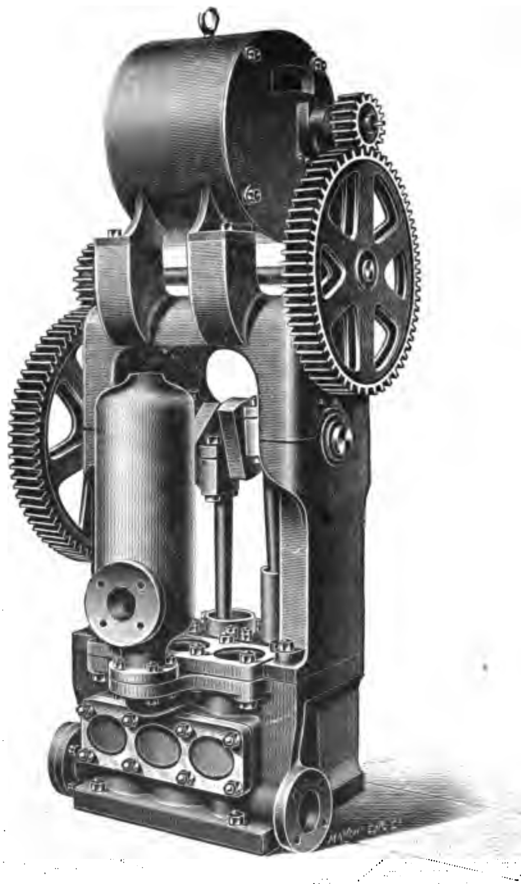


FIG 134.—Electrically-driven Treble-ram Pump by Messrs. Frank Pearn and Co. Ltd.

TYPES OF RECIPROCATING PUMPS.

Fig. 135.

Electric and Power Pumps.—Fig. 134 illustrates a treble-ram pump by Messrs. Frank Pearn and Co. Ltd., of West Gorton, Manchester, driven by an electric motor

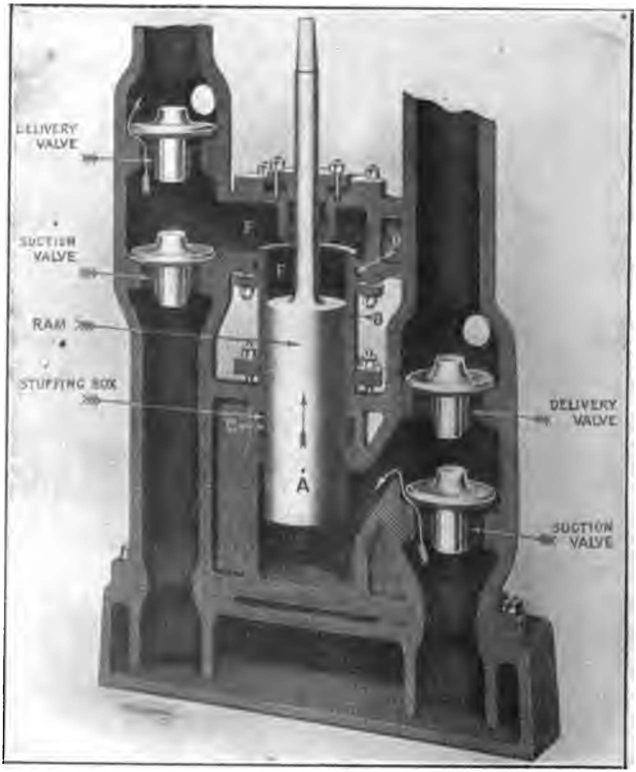


FIG. 136.—Pearn's Patent Packing for the Ram of Double-acting Pumps.

which is mounted at the upper end of the pump casing. The pump gearing is of cast iron, with machine-cut teeth; the motor pinion is of raw hide to ensure silent running.

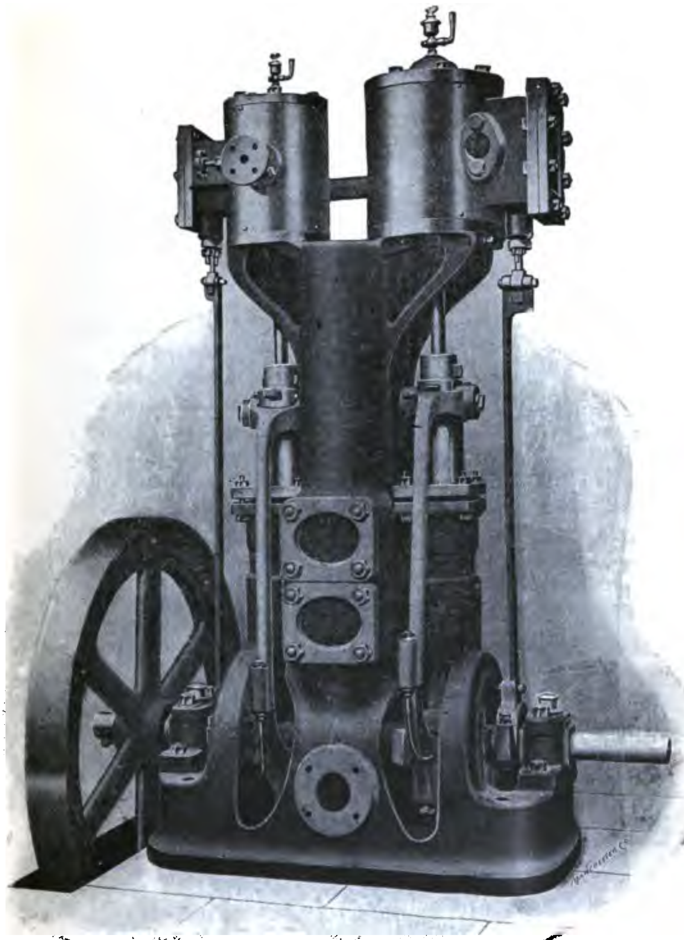


FIG. 137.—The "Manchester" Compound Steam Pump.

Fig. 135 represents a horizontal double-acting ram pump by the same makers. There is only one ram, but as it works in a divided barrel, as shown, the double action is obtained. The makers state: "The gearing is of cast iron, machine moulded, and the countershaft is fitted with flywheel and driving and loose pulleys; when the motive power is taken from line shafting running at not more than 60 revolutions per minute the gearing is not necessary, but when the speed exceeds 60 revolutions per minute the gearing as shown in the illustration is necessary."

Fig. 136 is a sectional illustration of Pearn's patent method of packing the ram of a double-acting pump of the vertical type well known as the "Cameron." The ram A is of the type used in the divided-barrel double-acting pump previously referred to, but made much shorter, as it works through one stuffing box only instead of two. The sleeve or barrel B is arranged as shown, between the lower pump chamber and the upper chamber F. The barrel has a flange attached to it, so that it may serve as a gland for the stuffing box C; as such combined gland and barrel is guided at its upper end, the makers state that it cannot be screwed down unevenly. A joint is made at the upper end by the ring D.

The Manchester Compound Steam Pump.—An example of this pump or pumping engine is illustrated at fig. 137. The ram barrels, valve boxes, delivery air vessel, and bottom covers of steam cylinders are all cast in one piece, providing a compact and strong combination with the working parts very accessible. The valves and valve seats are of gun-metal, each having a separate door for access. It is well adapted for boiler feeding and all general purposes.

CHAPTER XVIII.

CENTRIFUGAL PUMPS: TURBINE PUMPS.

CENTRIFUGAL pumps have received much attention during the last few years. The improvements of Appold, more than half a century ago, established the position of the centrifugal pump as a most suitable machine for raising large quantities of water against a low head or pressure. Modern improvements have made it not only possible, but commercially advantageous, to employ centrifugal pumps on high lifts up to, and even beyond, 1,000 ft.

The centrifugal pump has been said to be the converse of the turbine water-wheel, and its development described as analogous to that of the steam turbine, as both were pioneers in their respective fields and both were abandoned in favour of reciprocating machines. Such abandonment in the case of the pump was no doubt largely due to lack of a clear appreciation of the principles of its action, whilst with the steam turbine constructional difficulties required a man with the persistence, energy, and ability of Parsons to overcome them.

The recent advances in centrifugal pump practice have been brought about in the main by the necessity pressed home upon pump makers of producing machines capable of direct connection to an electric motor. The employment of toothed gearing is always to be avoided on pump service whenever possible, but when a slow-running machine, such as a reciprocating pump, is to be driven by a quick-running motor, such as an electric motor, speed-reduction gearing is in most cases an absolute necessity. In Germany and elsewhere quick-running short-stroke reciprocating pumps, styled "express pumps," have been put forward as a solution of the problem. But it is self-evident that as centrifugal pumps and electric motors are both rotary machines and require to be run

at high rotative speeds, they are eminently adapted to be coupled together for direct driving. Hence in this country, and also in Switzerland and in America, engineers have worked in such a direction rather than in the way of providing for the slow running of an electric motor and the quick running of a plunger or reciprocating pump, in the endeavour to reconcile mutually antagonistic conditions.

In a reciprocating pump the water is raised to the required height or discharged at the required pressure by virtue of the imposition thereon in the pump cylinder, through the medium of a slow-moving ram or plunger, of the necessary intensity of pressure. Thus, if the water has to be discharged from the pump at a pressure of 200 lb. per square inch, it will be necessary to impose a pressure of that amount per square inch (or a little more) upon the ram or plunger. In the expressive words of the old mechanics, we may say that the work is done by a large force (or pressure) moving through a small space or distance.

With a centrifugal pump, the moving part, the fan or impeller, imposes very little pressure, but a high velocity upon the water. We may say, therefore, that in this case the work is accomplished by a small force moving through a large space or distance. This method of imparting the necessary energy to the water involves the provision in the pump of means for the conversion of the velocity energy or kinetic energy into pressure energy or potential energy, in order, that the static head of water or resistance against which the pumping is effected may be overcome. It has been well said that the problem of the centrifugal pump designer is to so proportion the parts of his pump that it will pick up the water from rest, bring it to the high velocity required by the head pumped against, and then allow it to come to rest again, and to so do this that there shall be very little internal friction or loss by leakage or slip. In the ordinary or low-lift centrifugal pump the conversion of the velocity energy into pressure energy is effected mainly by the arrangement of

the portion of the pump casing surrounding the fan or impeller as a volute or spiral chamber, wherein much of the velocity of the water is converted into pressure. An additional space or chamber, known as the whirlpool chamber, is sometimes provided around the volute.

Every additional foot of head against which a centrifugal pump has to deliver involves a corresponding increase in speed, such increase being governed by the relation existing between velocity and head, as represented by the formula—

$$h = \frac{v^2}{2g},$$

h being the head in feet, v the velocity in feet per second, and g the rate of acceleration due to gravity. It must be remembered when applying this formula that the impeller, according to the shape of its blades, may impart, or tend to impart, a velocity to the water in more than one direction, so that in bringing the stream to rest the resultant pressure energy will be greater than if it had a velocity in one direction only. Thus, with radial blades or vanes, h would have just twice its value given by the formula.

As increase of lift means increase of velocity, it is clear that a high lift would involve either an excessively high rate of rotation or an impeller of very large diameter, if the work had to be accomplished with a centrifugal pump of the old form, having but one impeller. Hence the practice arose of employing a series arrangement of pumps, comprising a number of impellers fixed on the one shaft, with a separate casing for each and a pipe connection between the casings. Thus the water, after leaving one pump chamber under the pressure due to the conversion of the kinetic energy received from the impeller in such chamber, flows into the succeeding chamber, where it receives a further supply of kinetic energy, which is likewise subjected to conversion into pressure energy. It follows that the final pressure of the water will be (disregarding all losses) the pressure due to that obtainable

from the one impeller, multiplied by the number of impellers.

But a series arrangement such as aforesaid must generally be wanting in efficiency through the setting up in

Fig. 138

the water of eddy currents and internal commotions or shocks during the numerous conversions of the energy

from one form to the other. Thus there came into being the high-lift centrifugal, known as the turbine pump, in which the series of impellers mounted on the one shaft are arranged in a single casing, divided internally into a number of inter-communicating compartments, whilst diffusion vanes, guide rings or discs, or their equivalent, are employed to divert the water discharged from the impeller and reduce its velocity, for the conversion of the same into pressure.

To Messrs. Sulzer Bros., whose principal works are at Winterthur, Switzerland, and whose London house is at 34, Norfolk Street, Strand, W.C., engineers are much indebted for the active part taken by that firm in pioneering and developing high-lift centrifugal or turbine pumps.

The first Sulzer high-lift centrifugal pump was exhibited at the Geneva Exhibition, in the year 1896; it had but one impeller, was coupled direct with an electric motor, and running at 900 revolutions per minute delivered three cubic metres (660 imperial gallons) of water per minute against a head of 45 metres (148 ft.). In the following year Messrs. Sulzer Bros. supplied to the Geneva Waterworks a centrifugal pump having two impellers, each 1,100 mm. diameter (3 ft. 7¼ in.), and coupled direct to a 1,000 brake horse power electric motor. This pump, running at 540 revolutions per minute, delivers 22·5 cubic metres (4,950 imperial gallons) per minute against a head of 140 metres (460 ft.). At the city of Milan Waterworks, Sulzer high-lift centrifugal pumps, running at 925 revolutions per minute, deliver six cubic metres (1,320 imperial gallons) per minute against a head of 52 metres (170 ft.). For supplying the cooling water during the building of the Simplon Tunnel two Sulzer high-lift centrifugal pumps were employed, each of which delivered 4·8 cubic metres (1,056 imperial gallons) per minute at a pressure of 22 atmospheres (312 lb.) when running at 1,050 revolutions per minute, and required 325 horse power. When coupled in series the pumps could produce a pressure of 44 atmospheres (625 lb.) At Kom-Ombo, below Assuan, near the first Nile cataract, single impeller pumps, with spiral casing and running at 110 revolutions per minute, deliver

FIG 139.

as much as 200 cubic metres (44,000 imperial gallons) to a height of 22·4 metres ($73\frac{1}{2}$ ft.), requiring 1,250 horse power. The efficiency in this case is given as exceeding 80 per cent. It will be observed, however, that the actual horse power represented by the stated duty is

$$\frac{44000 \times 10 \times 73{\cdot}5}{33000} = 980.$$

As the horse power required is given as 1,250, the efficiency would appear to be

$$\frac{980}{1250} \times 100 = 78 \text{ per cent.}$$

Fire engines have been built consisting of Sulzer high-lift centrifugal pumps directly driven by a motor, which can be connected, by means of flexible cables and plugs, with the electric supply main.

In 1898 Messrs. Sulzer Bros. supplied a group of three high-lift four-stage pumps, with impellers 500 mm. diameter ($19\ {}^{11}/_{16}$ in.), for the silver mine at Horcajo, in Spain. Each pump is capable of forcing 1,000 gallons per minute against a head of 500 ft., but they are arranged at different levels in the mine, the lowermost delivering to the intermediate pump, whilst the latter forces to the upper pump. By this series arrangement 1,000 gallons is raised per minute against a total head of 1,500 ft. Still greater heads have been overcome by a similar series arrangement of high-lift pumps. But the arrangement now more generally adopted for deep-mine service by Messrs. Sulzer Bros. consists in placing the pumps in series at the one level; the head may then be said to be overcome in a single lift. The first typical plant of this kind was installed at the Victor Mine, at Rauxel, Westphalia. The guarantee for the two four-stage high-lift pumps, which are placed in series, was for a delivery of seven cubic metres (1,540 imperial gallons) per minute against a total head of 524 metres (1,720 ft.) at 1,040 revolutions per minute, each pump requiring 570 brake

Fig. 140.

horse power. The actual horse power represented by the stated figures is

$$\frac{1540 \times 10 \times 1720}{33000} = 803.$$

But as the makers stipulated for 570 horse power for each pump, or 1,140 horse power for the work to be done by both, they evidently calculated on an efficiency of 70 per

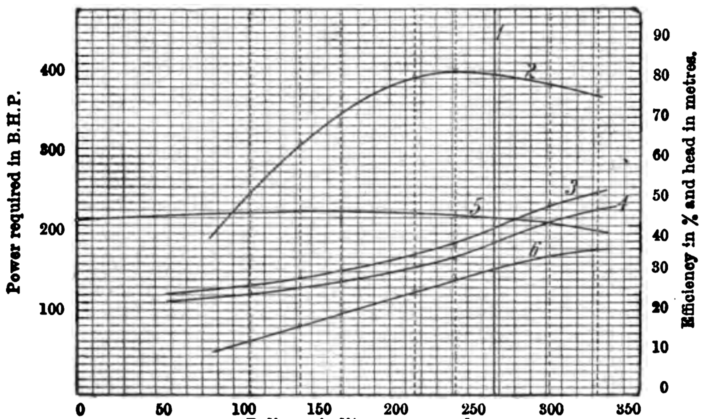


FIG. 141.

1 = Normal output.
2 = Efficiency of pump.
3 = Power required by motor in H.P.
4 = Power required by pump in H.P.
5 = Head.
6 = Theoretical output of pump.
Speed, about 1025 revs. per min. constant.

cent for their machines. On official tests the efficiency of the pumps proved to be 76 per cent. It is of interest also to note that the efficiency of pumping set (pumps and motors) was $71\frac{1}{4}$ per cent, and the total efficiency of plant (steam engines, pumps, and motors, also cable losses), 59 per cent.

The single-lift series arrangement of pumps also permits of what may be described as pumping in parallel for quan-

tity. To accomplish this, each pump has simply to be set (by adjustment of gate valves in the respective pipes) to draw water independently from the sump. It should also

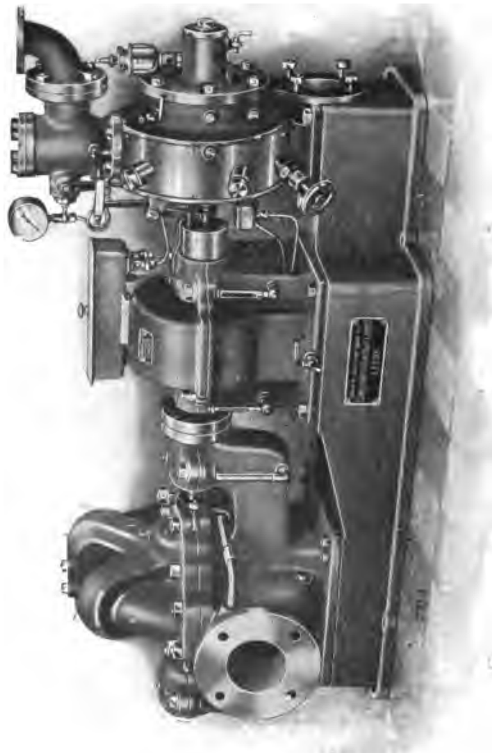


FIG. 142.

be observed that instead of employing a separate motor for each pump of a series arrangement, the two pumps may be coupled on either side of the motor and the delivery

pipe of the one pump connected to the suction nozzle of the other.

Fig. 138 illustrates a six-stage Sulzer high-lift centrifugal pump adapted for belt drive. Fig. 139 represents another six-stage pump by the same makers coupled direct with a Schuckert asynchronous motor of 225 brake horse power normal output, taking current of 1,000 volts and 50 cycles. The motor has no slip rings, and is provided with water-cooled ring bearings. With 206 brake horse power, and running at 1,455 revolutions per minute, the capacity of the pump is given at 2·52 cubic metres (554 imperial gallons) per minute against a head of 255 metres (837 ft.). The horse power represented by the work done amounts to

$$\frac{554 \times 10 \times 837}{33000} = 141.$$

This gives an efficiency of

$$\frac{141}{206} \times 100 = 68 \text{ per cent.}$$

For connecting up pump and motor Messrs. Sulzer employ an elastic or flexible coupling, one half of which is provided with pins while the other has corresponding apertures lined with rubber.

Figs. 140 and 141 are illustrations (which, as with figs. 138 and 139, were kindly supplied by the makers) respectively representing a Sulzer sinking pump and the curves showing the results obtained. The pump was built for a delivery of 16 cubic metres (3,520 imperial gallons) per minute to a manometric head of 45 metres (147 ft.), with a speed of 1,025 revolutions per minute. At the guaranteed output the efficiency attained is given as 83 per cent, and, as shown by the curve, this increased to 84 per cent under somewhat different conditions of working.

Fig. 142 is an illustration showing a 15 horse power De Laval steam turbine and centrifugal pump by Messrs. Greenwood and Batley Limited, of Albion Works, Leeds; fig. 143 is a view with the pump-case cover removed to show the interior construction of the pump. The makers state

the De Laval patent centrifugal pumps are made in standard sizes for a very wide range of lifts. For use as boiler-feed pumps they are made for working against

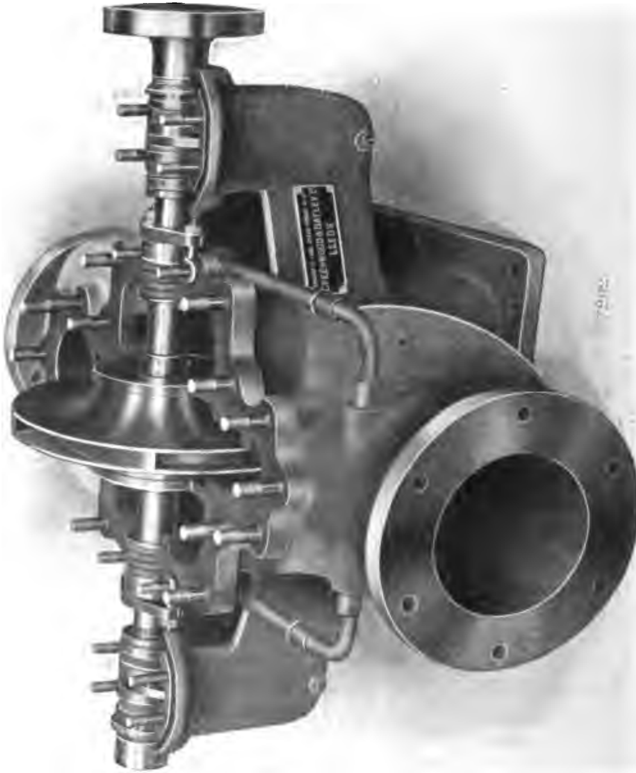


FIG. 148.

pressures of 250 lb. to 300 lb. per square inch. Whilst the pumps are specially designed for working in combination with high-speed steam turbines and electric motors, they are also constructed for lower speeds suitable for belt

driving or for connection to slow-speed engines or motors. Referring to what is described as the special characteristic of the De Laval centrifugal pumps, viz., their high speed, the makers state that such speed "is greatly in excess of anything previously known in connection with centrifugal pumps, and is a mechanical problem which has been solved partly by an improvement in the construction of the wheel (impeller) and packing boxes of the pumps, including their lubricating arrangements, and partly by paying the greatest attention to the execution of all the movable parts. The high rate of speed has enabled the diameter of the pump wheel to be reduced, and consequently the frictional resistances of the pump have been greatly lowered. The dimensions of the pumps are small, and their weight, as compared with their capacity, is low. All bearings are self-oiling and entirely separated from the packings, thus making it impossible for any oil to get into the water. The larger sizes of steam turbine pumps have two driving shafts, with one pump coupled to each shaft; by connecting the two pumps in parallel a large quantity of water is delivered against a low head, whilst by connecting them in series a smaller quantity is delivered against a greater head." In one example given in the makers' standard lists a speed of 4,000 revolutions per minute is specified; this particular pump, driven by a 5 brake horse power steam turbine, delivers 90 gallons per minute against a total head of 90 ft.

As is well known, the pressure in a centrifugal pump cannot be increased to a dangerous extent (as may happen with some piston pumps), through such a cause as the accidental stoppage of the delivery through the closing of a gate valve or otherwise. Considering this point with regard to the De Laval centrifugal pumps, Messrs. Greenwood and Batley state: "Should the delivery pipe be shut off during the time the pump is working at full speed, the water pressure could not rise beyond the so-called 'centrifugal pressure,' which, as a rule, is about 25 per cent above the ordinary working pressure."

Fig. 144 represents a high-lift centrifugal or turbine pump (electrically driven), by Messrs. W. H. Allen, Son, and Company Limited, of Queen's Engineering Works, Bedford, who give the following particulars: "The pump illustrated is of the high-lift turbine type, the impellers being mounted upon the same spindle and arranged to work in series, so that the total lift of the pump is the sum of the head generated by each impeller. The pump in question is suitable for a lift of about 300 ft., discharging 600 to 800 gallons per minute, when running at 1,200 to 1,400 revolutions per minute. The pump casing is constructed of cast iron and the spindle of nickel steel, the impellers and guide vanes being of special high-tension bronze. The water, on leaving the guide vanes, is led to the eye of the next disc by passages which have been carefully arranged so as to cause the water to move with as little friction as possible. The last impeller of the series delivers its discharge through an annular whirlpool chamber or diffuser, from whence it passes into the delivery pipes. A balancing piston is fitted, being so adjusted as to neutralise the end thrust from the pump, and a small ball-thrust bearing is also provided to take up any small unbalanced end-thrust which may arise during working. All the bearings of the pump are of the self-oiling type, so as to enable the pump to run with the minimum of attention. The casing is provided with all the necessary accessories in the way of air vents to clear the pump of air when starting up. This pump was tested at our works at Bedford, and every precaution was taken to make the test a true account of the performance of the pump. For this and similar work we have recently installed two 'Venturi' water meters in our testing department, carefully calibrated, which render it possible to read the discharge of the pump in gallons per minute at any instant directly by means of a mercury gauge. The efficiency of the pump was 75 per cent when on a lift of about 300 feet. Great care has been taken in the design of these pumps to ensure smooth running and to eliminate every possible cause of vibration, and with this end in view the greatest care has been taken in the balance

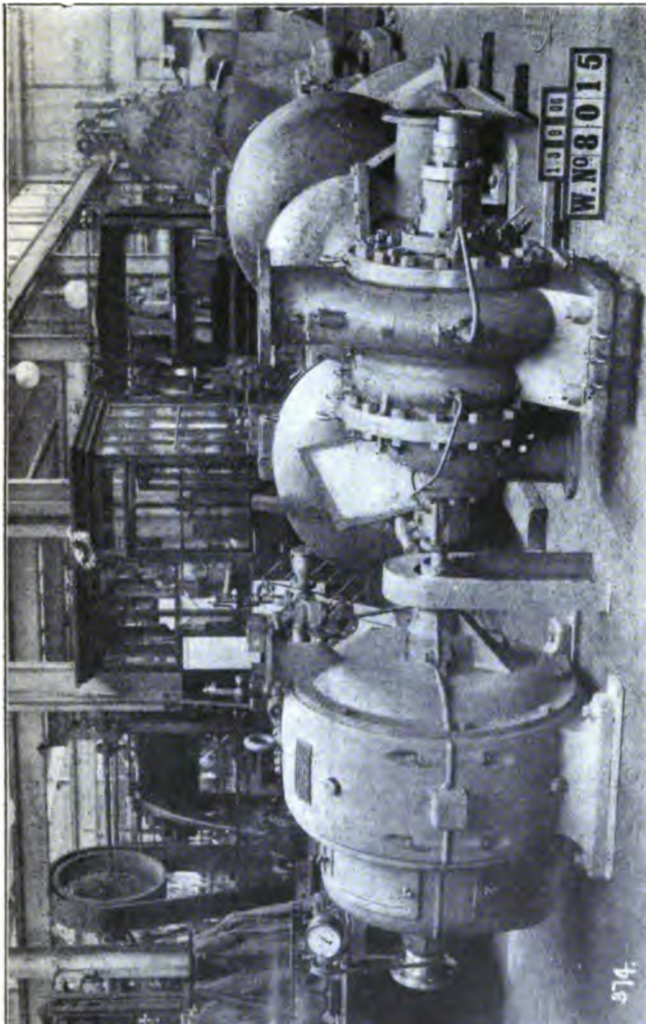


FIG. 14[illegible].—High-Lift Centrifugal or Turbine Pump, by Messrs. W. H. Allen, Son, and Co. Ltd.

and design of the spindle and the rotating parts carried by it."

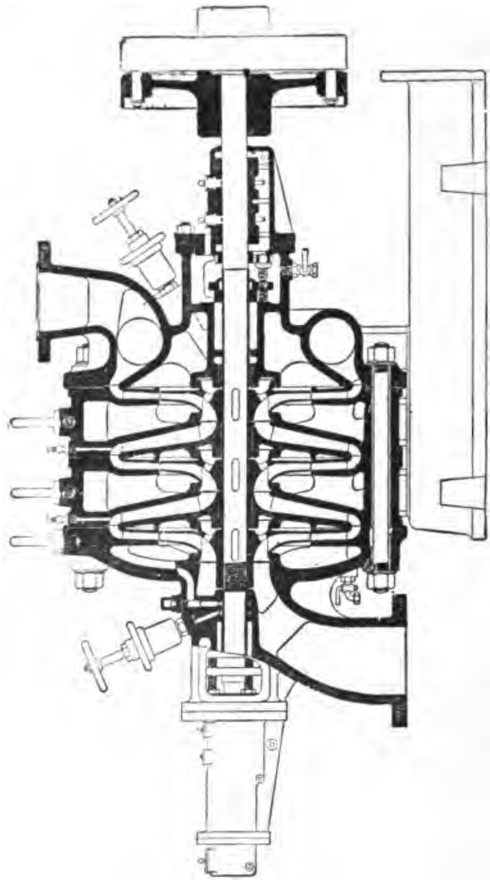


Fig. 145.—Section of a Worthington Multi-Stage Turbine Pump.

Fig. 145 is a general section of a Worthington multi-stage turbine pump by the Worthington Pump Company Limited, of 153, Queen Victoria Street, London, E.C.

The following is from the makers' description: "After leaving the suction opening, the water enters the impeller through an annular opening of ample area, and, directed by the vanes, is discharged at the desired velocity at the periphery of the impeller. The velocity of the water, which is highest at this point, is now gradually reduced, by means of diffusion vanes designed in such a manner as to convert the velocity head of the water into static pressure with the minimum frictional loss. If the pump has only one impeller, the water, on leaving the diffusion vanes, passes into the discharge casing and then to the delivery pipe. Fig. 145, however, shows a multi-stage pump, and in this case the water leaving the diffusion vanes of the first impeller passes, in the most direct manner possible, through the intermediate casing into the suction opening of the second impeller, this operation being repeated as often as is necessary to procure the final delivery pressure. The turbine pump casing is of the best quality special cast iron or cast steel, according to the working pressure required. The casing in multi-stage pumps is made up of separate sections, accurately machined and bolted together, this method of construction enabling the pump to be dismantled in the shortest possible time when any examination of the internal parts is necessary. The impellers are of special hard phosphor-bronze, carefully machined, and polished all over. In order to avoid end-thrust a suitable number of holes are drilled through the impeller boss. Any further tendency to produce unequal pressure on either side of the impeller is entirely eliminated by an arrangement adopted in the design of the impeller and the diffusion vanes; this arrangement, while simple, is perfectly automatic and self-adjusting. The diffusion vanes, through which the water passes after leaving the impeller, are made of hard phosphor-bronze, machined all over, and firmly secured to the pump casing. In order to obtain the maximum strength, and to guard against corrosion, the shafts in all our turbine pumps are made of high-percentage nickel steel, accurately turned and polished. The pump bearings in all but the smallest

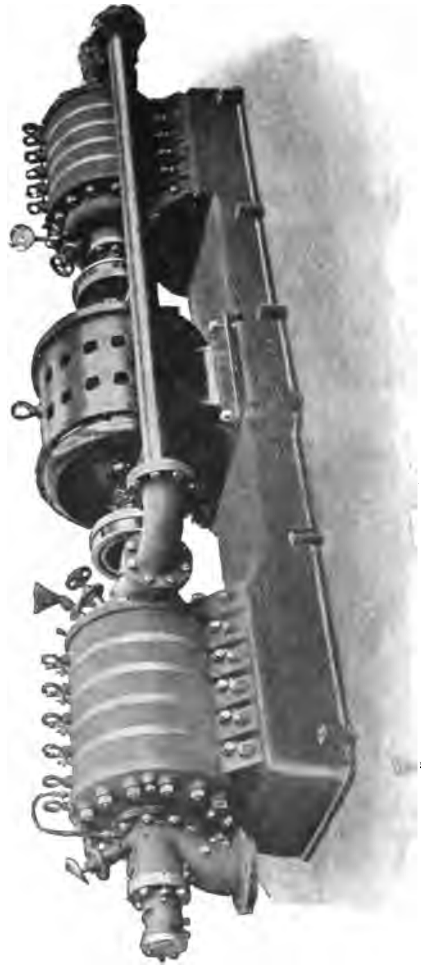


FIG. 146.—Worthington Twelve-Stage Turbine Pump driven by Electric Motor.

sizes of turbine pumps are arranged entirely separate from the stuffing boxes. This arrangement prevents all chance of grit or foreign matter entering the bearings. The bearings are of the 'ring-oiled' type, specially constructed for high-speed running."

Upon the question of the speed of a turbine pump and the relation of capacity to head, the Worthington Pump Company, after mentioning the limitation of speed necessary if the water is gritty, in order to prevent too rapid wear of the internal parts of the pump, make the following statement: "It is not possible to fix any hard-and-fast rule about the relation of capacity to head, but, generally speaking, at ordinary speeds, to obtain the very best results with a pump of moderate price, the number of gallons per minute should be about equal to the number of feet in the total head. To take an example: A pump designed to deliver 500 gallons per minute against a head of 500 ft. would give an efficiency of not less than 72 per cent, while the best efficiency possible with a pump designed to deliver 50 gallons against the same head would only be about 60 per cent. In the latter case, not only would the efficiency be reduced, but the price of such a turbine pump would in all probability be more than that of the best type of three-throw pump. It goes without saying that as the quantity increases so the efficiency will increase, and from the two cases cited above it will be seen that where the head is greatly in excess of the quantity a reciprocating pump will most probably give the best results. Each case, however, must be considered on its merits, and there will be many instances in which the lower first cost, economy of space, lower cost of upkeep, and the general convenience in arrangement of turbine pumps, will give them a distinct preference over the reciprocating type."

Fig. 146 is an illustration of a Worthington twelve-stage turbine pump with electric motor, the latter being placed between a pair of pump casings, each having six impellers.

Respecting the speed of pumps and the type of electric motors for driving same, the Worthington Pump Company

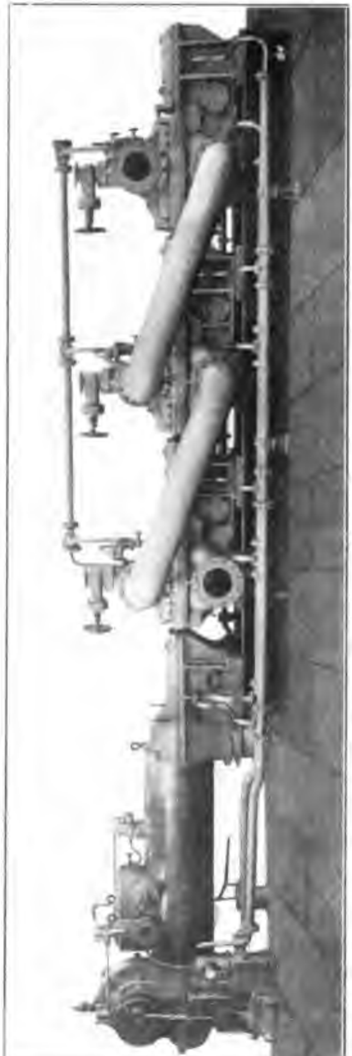


FIG. 147.—Set of Centrifugal Pumps directly driven by a Parsons Steam Turbine.

say: "In most circumstances it may be said that motors of the alternating-current type are preferable, mainly on account of their adaptability to high speeds. Although there is no standard speed for turbine pumps, 1,440 to 1,470 revolutions per minute may be taken as being suitable in most cases for pumps requiring motors above 10 B.H.P. When the power required is less than this a much higher speed may be adopted. Under normal conditions the ordinary induction motor is perfectly suitable for driving our turbine pumps, owing to the fact that the current necessary to start the pump will not exceed the full-load current. High-speed direct-current motors having been for some time the subject of special design by the principal electrical manufacturers, the speeds mentioned above as being suitable for alternating-current motors may now be adhered to without difficulty when direct-current motors are used. When vertical motors are supplied, they should be provided where possible with a suitable ball-thrust bearing on the motor spindle capable of carrying not only the weight of the armature, but also of the intermediate shafting and pump impeller."

Fig. 147 represents a set of centrifugal pumps directly driven by a Parsons steam turbine. This plant, constructed by Messrs. C. A. Parsons and Co., of Heaton Works, Newcastle-on-Tyne, is capable of dealing with 1,100 gallons of water per minute at 760 ft. head as a normal duty, and 1,170 gallons at 920 ft. head as a maximum duty.

Some years ago Messrs. Mather and Platt Limited, of Salford Ironworks, Manchester, introduced a multiple-chamber centrifugal pump, known as the "Mather-Reynolds patent high-lift turbine pump." This pump had one or more sets of vanes or impellers, each running in its own chamber, but upon a common shaft. With this pump the makers state that they "were able to deliver to a height of 150 ft., a result never before obtained with a centrifugal pump." With reference to their turbine pumps as now made, Messrs. Mather and Platt give the following description: "In the improved type of patent turbine pump, as now made by us, axial thrust is eliminated; the

water enters the revolving wheel axially, traverses the curved internal passages between the vanes, and is discharged tangentially at the periphery into a stationary guide ring of special construction; this conveys it to the annular chamber in the body of the pump, where the velocity head imparted to the water by the wheel is converted into pressure head. From this chamber the water is finally discharged into the pipe lines, or, if the pump be a multiple one, into the second and subsequent chambers. A special feature of this pump is the provision of the stationary guide ring mentioned above: this is

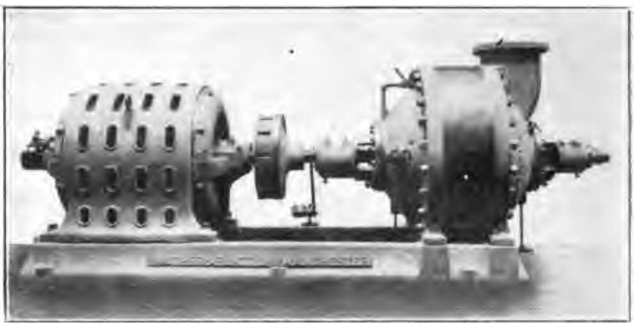


FIG. 148.—Electrically-driven Centrifugal Pump, by Messrs. Mather and Platt Ltd.

fixed concentric with the revolving vanes, and, owing to its design, enables the conversion of velocity into pressure head to be carried out in a much more perfect manner than is possible in the case of any other centrifugal pump; thus, not only is the possible height of lift, but also the efficiency of the pump greatly increased. Another point of interest is that with the special design of guide passages in these pumps the water is nowhere forced to undergo a sudden change of direction or to meet with a sudden difference of cross-section in the passages."

Figs. 148 and 149 illustrate examples of pumps by Messrs. Mather and Platt Limited. Fig. 148 depicts a two-chamber pump, driven by a three-phase induction motor,

for raising 3,820 gallons of water per minute against a total head of 300 ft. The motor works at 2,200 volts and 63 cycles per second. The speed of pump and motor is 740 revolutions per minute. The plant was supplied to the Montreal Water and Power Company. An official test gives an efficiency of 70·8 per cent. A second plant for the same company delivers 4,500 gallons per minute against the same head. In the latter case the pump is coupled at one end to an electric motor and at the other end to a steam engine. This arrangement involved a much lower speed

FIG. 149.—Electrically-driven Centrifugal Pump (assisted by turbine), by Messrs. Mather and Platt Ltd.

for the pump—namely, 335 revolutions per minute—and hence the number of chambers and impellers had to be increased from two to six. Fig. 149 shows a single-chamber pump, one of three supplied by Mather and Platt to the Newcastle Corporation for lifting water from the river Tyne and circulating it through the condensers in the power station of the electricity works. The pump illustrated is for delivering 2,500 gallons per minute against 100 ft. to 117 ft. head, according to the height of the river, when running at 700 revolutions per minute. It is driven by a direct-coupled electric motor assisted by

a water turbine actuated by the return water from the condensers to the river.

In another example of a high-lift centrifugal pump by Messrs. Mather and Platt the pump efficiency on test, at 1,000 gallons per minute against a total head of 320 ft., is given at 75 per cent, and the combined efficiency of the pump (four-stage) and motor (three-phase) 69·5 per cent.

CHAPTER XIX.

The Air-lift System of Pumping.

For the raising of water from deep wells it has long been the practice to employ three-throw or other reciprocating pumps, which are fixed in the well within suction distance of the water, and operated by an engine or power mechanism arranged at the surface. But in what is known as the "air lift" we have an alternative system, and one that is rapidly coming into general use. An air-lift plant, so far as the well itself is concerned, consists merely of two pipes, the larger one for water and the smaller for air. The air pipe may be fitted within the water pipe or placed alongside of it. Both pipes must dip considerably below the surface of the water—in fact the dip, or immersion, must exceed height through which water has to be raised—and the bottom of air pipe enters the water pipe in order that compressed air may be forced into the latter. The bottom of the water pipe is open. The compressed air passes up through the water pipe, carrying with it a quantity of water. Under some conditions of working the air and water form an emulsion which is discharged with some force through the water pipe or rising main. But under more normal conditions, when the apparatus is fairly under way, the air ascends the rising main in large bubbles, the water forming a corresponding series of layers interposed between the air bubbles. An American writer has thus

described the action: "Air being admitted or forced into the water pipe, a large bubble is formed about the orifice of the air pipe which fills the water pipe (*i.e.*, occupies the entire cross section), and forms what may be termed a piston of air. The pressure of the air when escaping from the air pipe is sufficient to slightly overbalance the weight of the column of water in the water pipe, and a quantity of water equal to volume of air admitted flows out at the top. The bubble of air immediately begins to rise through the water, and as it rises it continues to expand, due to the constantly decreasing weight of the column of water above it. The expansion of the air, and the consequent increase in volume, displaces a corresponding volume of water at the top, until finally the air reaches the outlet, where it escapes, carrying a quantity of water in the form of spray. The next bubble reaches the top in the same manner, and raises a similar quantity of water. These air bubbles form in rapid succession, thus dividing the column of water in the water pipe into alternate layers of air and water, each bubble or piston having a layer of water above it. It will be seen that after the column of water has once become divided into alternate layers of air and water, the air admitted at the bottom does not have a solid column of water to sustain or balance, consequently the pressure of air may be reduced after the pump begins to work properly, and the volume of air thus reduced accordingly, which lessens the work of the compressor. The pressure of air required to operate the pump after it is once started is considerably less than the pressure of water due to the head, while the air pressure required to start the pump is slightly greater than the water pressure corresponding to the head, and is generally equal to the water pressure per square inch, plus 5 lb., or air pressure $= 0{\cdot}433 \times H + 5$, where H is the head or height in feet of the column of water in the water pipe, measured from the point where the air pipe enters the water pipe."

The first application of the air lift is said to have been by one Crockford, in 1846, who employed it for the pur-

pose of raising petroleum from Pennsylvania wells, but for several years after that there appears to have been little or no further application of the principle.

Messrs. Le Grand and Sutcliffe, of Magdala Works, 125, Bunhill Row, London, E.C., make an air-lift pump described as "The Multex," or multiple-expansion type, and of which they furnish the following particulars: "The rising main of the ordinary air-lift pump is of equal diameter throughout, and experiments have shown that layers of air rise in the main alternately with water, and as the layer of air rises in the main the pressure above it, of course, grows less, at the top the pressure being only that of atmosphere. If it can do so the air will expand gradually on its way up, until at the top the length of tube that it occupies will be several times greater than that which it occupied at the bottom; or, in other words, much of the energy of the air is absorbed by giving the water a rapid motion, which is not required. This point has been very carefully studied by Mr. Joseph Price, of Messrs. Le Grand and Sutcliffe, who has for a long time been investigating the matter, with the result that the firm have introduced a form of air-lift pump in which the rising main is gradually widened from the bottom towards the top. By this arrangement it will be seen that the air can expand laterally, and the layers of water will have only a slightly higher velocity at the top of the pipe than at the bottom. The energy expended in giving a higher momentum to the water as it rises is thus saved."

The air lift depends very largely for its successful operation upon the ratio of depth of air pipe immersion to height through which it has to be lifted; the relative areas of rising main and air pipe is also of much importance. It would appear that about 60 per cent of immersion to 40 per cent of lift is generally found to give the best results, but such figures must not be taken as applicable in all cases. The immersion may apparently range between 55 and 80 per cent. The best ratio of the area of the air pipe to water pipe or rising main has been given by one writer as about 0·16.

Some very interesting and useful information relating to air lifts will be found in a paper by Mr. James Kelly, published in Vol. CLXIII. (May, 1906), of "Proceedings of the Institution of Civil Engineers." Mr. Kelly's paper gives the results of his experience with and tests on such pumps at the works of the United Alkali Company Limited, at Preesall, Lancaster. The maximum efficiency he records is 38 per cent, such efficiency representing the ratio between the actual work done in water raised to the work indicated in the steam cylinders of the air compressor. The cited efficiency value was obtained when delivering against about 100 ft. head, with a mean air pressure of 110 lb. per square inch; the delivery was at the rate of 636 gallons per minute, with an air consumption of 307 cubic feet per minute. The indicated horse power in steam cylinders of air compressor was 51, and the actual horse power in water raised 19·39. Such an efficiency is a very good one for an air lift. It will appear low when compared with the efficiencies obtainable from large triple-expansion pumping engines working under favourable conditions, but it must be remembered that air lifts are employed where such conditions do not prevail.

The Worthington Pump Co. Ltd. thus express the advantages of air lifts: "The submerged parts are subject to no wear whatever, and, except for corrosion, are practically indestructible even when working in the gritty water so fatal to deep well pump pistons. The air compressor and receiver, the only parts needing attention, are at the surface, and readily accessible. Our experiments have been extensive and systematic, and we have determined conclusively that when proper submergence of the air nozzle can be assured, the yield of a well previously fitted with a deep well pump is doubled or trebled by the use of the air lift. The sparkle and life imparted to the water raised, and the cooling effect of the expanding air, make this method peculiarly suitable for water works purposes; while, owing to the absence of pump rod and plunger friction and the superior steam economy of the flywheel compressor, better duty can be

obtained than with the old style pump. When warm water is to be raised, its heat, instead of being a disadvantage, is a source of increased economy. A certain minimum ratio between submergence of the nozzle and

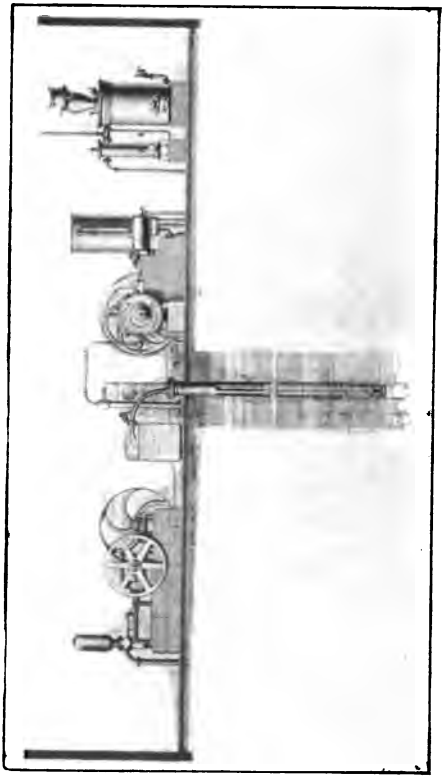


Fig. 150.

height of lift is, however, essential to the successful working of an air lift."

Fig. 150 illustrates an air-lift pumping plant by Messrs. C. Isler and Co., of Artesian Works, Bear Lane, South-

wark Street, London, S.E. The set comprises a combined air lift and surface pumps run by suction gas, and the particular plant illustrated is fixed at Bembridge, Isle of Wight. On the right of the drawing is shown the suction gas producer, with the filling hopper at the top, the starting fan on one side and the gas scrubbers on the other side. From the scrubbers the gas goes through an expansion box to the engine, the action of the latter when at work drawing the gas from the producer in the well-known manner. Bolted direct to the side of the gas-engine bed is the compressor. This has four single-acting cylinders set in the circular casing shown in the drawing, the four connecting rods being driven from a single crank pin. A solid crank disc, forged on the end of the engine shaft, carries this pin so that the air compressor and gas engine make one combined machine, with only two main bearings, the compressor having no bearing of its own. When starting, the air is released so that the compressor runs light. From the compressor the air passes through a combined receiver and oil extractor to the bore hole, and thence down the air-lift pipes to the footpiece, where it mixes with the water; air and water together are discharged through the delivery bend into the surface tank, where the air instantly separates from the water. From this tank the water is drawn by the three-throw horizontal-surface pump, and forced through the mains into the reservoir. The surface pump is belt-driven from a pulley on the flywheel side of the gas engine. The lift to surface by compressed air is about 80 ft., and the surface pumps deliver the water 170 ft. higher, making 250 ft. total lift. The makers give the water supply raised as 5,000 gallons per hour, and the cost for fuel at under one penny for the 5,000 gallons. Another plant on similar lines is fixed at Burnham Beeches Waterworks. From the figures for this case the lift to surface is 120 ft., and the lift above surface 160 ft., and the plant runs on 20 lb. weight of fuel per hour, or a cost of $\frac{1}{4}$d. per 1,000 gallons raised 300 ft., and over an average of six months' working the total cost comes to less than one-third of a penny per 1,000 gallons raised the total lift.

Semi-rotary Wing Pumps.

Fig. 151 is an external view, and fig. 152 a section representing one of the well-known Willcox semi-rotary wing pumps by Messrs. W. H. Willcox and Co. Ltd., of 23, Southwark Street, London, S.E. The pump illustrated is of the double-acting type, and consists, as will be seen, of a cylindrical casing containing a wing carrying the delivery valves, and adapted to be rocked or oscillated by the external lever. The lever shown is intended for

Fig. 151.

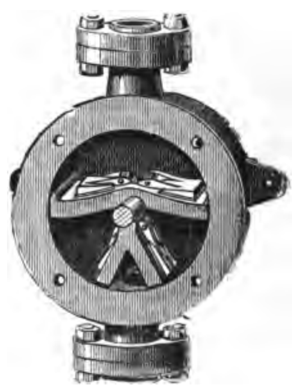


Fig. 152.

operation by hand, but the spindle on which the wing is mounted can, if desired, be fitted with a lever adapted to be reciprocated by power. Fig. 153 represents by two sectional views, taken at right angles to each other, the quadruple-acting semi-rotary wing pump (Abrahamson's patent) by the same makers. The pump is described as being "precisely similar in appearance to the double-acting wing pumps, the only change being in the internal mechanism." The body of the pump is divided into four chambers

instead of two, and is fitted with two suction valves at the bottom, and two retention valves at the top, the clacks or valves on the wings being dispensed with. The wing has two channels or ports which intersect one another, and place the chambers under them alternately in connection. The makers thus describe the action: "When the lever is pushed to the right the water is sucked in through the one valve, and falls into the chamber con-

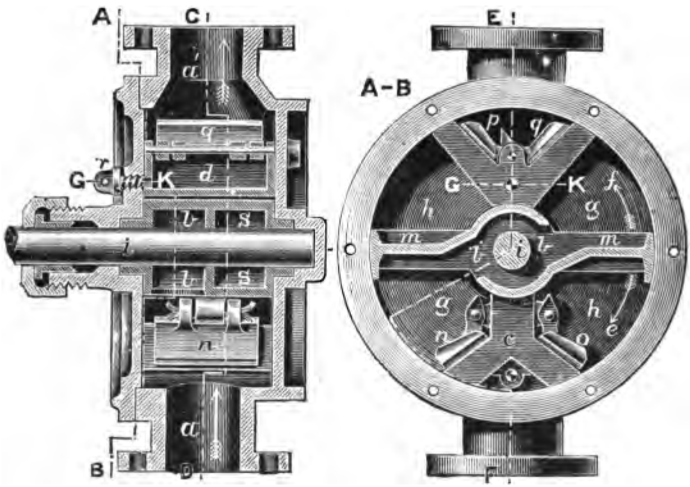


FIG. 153.

tiguous thereto, from whence it runs through the port into the next chamber. On moving the wing in the opposite direction both chambers are filled in the same way through the valve and the port, while the chambers on the other side are emptied by the port and the valve, etc." The advantages claimed include high capacity and compactness.

Both types of wing pumps may be fitted with ball or other valves to suit varying services.

Fig. 154 illustrates by two sectional views a pump by

Messrs. Willcox, and described by them as the "Record" convertible double-acting pump. It is adapted to be

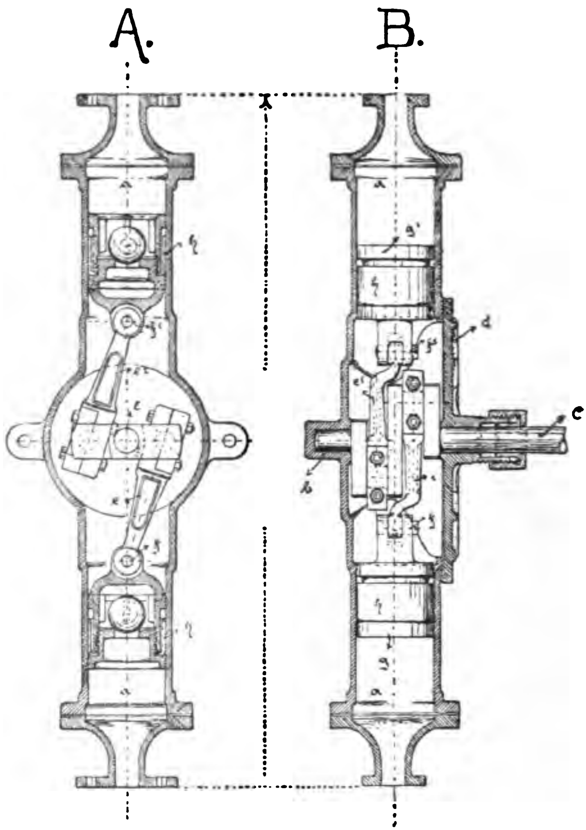


FIG. 154.

operated either by a semi-rotary or a rotary movement. As shown by the external view at fig. 155, the pump is

FIG. 155.

fitted with a hand lever, and is operated by a rocking movement of the latter. For the rotary action and power driving the lever has simply to be replaced by a belt pulley. Although described as a double-acting pump, it is actually an example of double delivery with single suction.

Fig. 156.

Another type of pump by the same makers, and adapted for working either by hand or power, is the "Willcox-Turner" patent pump, illustrated by figs. 156 and 157. This pump is described as representative of the makers' latest practice, and is but just ready for the market. The two pistons with which it is provided are each operated by a side rod connected, in the example shown, to the operating handle or lever. Each piston has two cup

leathers, placed back to back, "so that," the makers state, "one leather on each piston is always forcing, thus ensuring a good vacuum and no slip." The special advan-

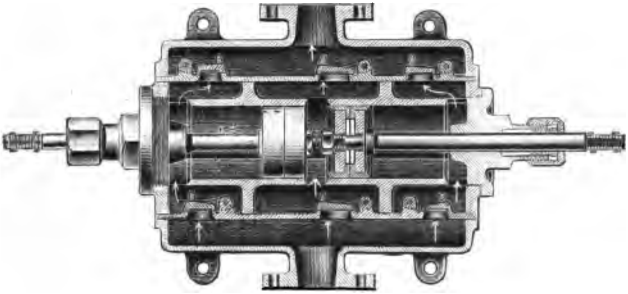


FIG. 157.

tage claimed is that "when working it is alway sucking and always delivering and there is an equal strain on the pump in whatever position the handle may be."

FIG. 158.

EJECTORS OR STEAM JET PUMPS.

An ejector or steam jet may be sometimes advantageously employed for the raising of water. Fig. 158 is an illustration of the "Penberthy XL-96" ejector or steam jet pump as made by Messrs. Willcox, whilst fig. 159 depicts an example of its application. The makers state that, whereas the lifting power (suction lift) of an ejector is usually seriously affected when the water has to be

delivered to a considerable elevation, making it necessary to place the ejector very near the water to be raised, they are able "to lift water 20 ft. with any steam pressure above 25 lb., and 25 ft. with pressure at 40 lb. or over. On pressures above 80 lb. steam must be throttled to get

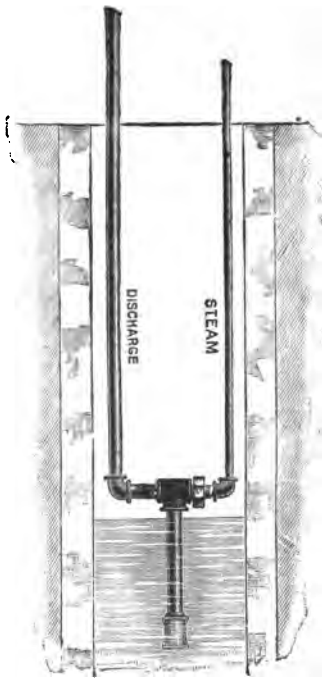


FIG. 159.

best results on a lift. Besides doing this, we elevate 50 ft. at 60 lb. pressure, and 65 to 75 ft. on pressures of 80 to 100 lb. by placing the ejector within 12 to 18 in. of water supply."

APPENDIX.

PATENTS RELATING TO CENTRIFUGAL AND TURBINE PUMPS.

No.	Year.	Name of Patentee and Subject Matter.
1446	1888	L. Vojacek. Arrangement of casing, delivery pipe, &c.
8119	,,	C. L. Hett. Suction pipe attachment.
17862	,,	J. E. Lawrence. Construction of casing.
1723	1889	H. A. Entwistle. Construction of suction pipe and blades.
2036	,,	W. Sayer. Arrangement of blades.
2603	,,	W. Beaumont. Arrangement of pump.
6421	,,	A. Brown. Priming arrangements, &c.
12136	,,	E A. Von Schmidt. Construction of suction branch and blades.
13833	,,	A. Robbins. Arrangement of bearings.
15041	,,	K. Leverkus. Construction of pump wheel.
16099	,,	P. Nezeraux. Arrangement of pump.
17505	,,	J. Martinez-Ancira. Arrangement of pump.
19636	,,	F. Pelzer. Adjusting outlet of centrifugal pump.
7456	1890	A. de Brouchere and L. Anspach. Controlling the motor driving centrifugal pump.
7915	,,	R. Anderson. Centrifugal pump for condensing system.
8020	,,	R. A. Lister and M. Pederson. Arrangement of pump.
10832	,,	J. P. Bayly (J. Edwards). Protecting pump casing, &c.
13677	,,	J. N. Paxman (J. Welman). Protecting pump casing, &c.
15755	,,	C. E. A. Rateau. Construction of pump.
20603	,,	J. A. Wade and J. Cherry. Priming arrangements.
6166	1891	J. C. Taite, T. W. Carlton, and R. M. Deeley. Centrifugal pump used for indicating speeds.
8026	,,	E. Seitz. Arrangement of pump
12610	,,	G. M. Capell. Form and arrangement of blades.
21376	,,	J. Mossop. Arrangement of pump.
5476	1892	M. R. Ruble. Arrangement of blades and outlet orifices.
7397	,,	Masson, Scott and Co. Ltd. and J. Taylor. Shape of casing and blades.
453	1893	A. Rateau. Arrangement of blades and deflecting surfaces.
5294	,,	C. L. Hett. Making impeller easily removable and preventing entrance of air through the stuffing boxes.
6368	,,	A. D. Ellis and W. B. Crichton. Making the parts interchangeable.
6758	,,	C. Herscher. Form of blades.

No.	Year.	Name of Patentee and Subject Matter.
7783	1893	G. Hanarte. Arrangement of outlet orifice, &c.
10470	,,	J. A. Wade and J. Cherry. Form of casing and blades.
18065	,,	E. J. Hawley. Form of protective lining and bearing.
19953	,,	G. M. Capell. Form and arrangement of blades.
21510	,,	R. J. Urquhart (P. F. Schmidt). Arrangement of blades.
483	1894	E. Seitz and R. P. Park. Protecting the bearings, balancing end thrust, &c
3723	,,	J and G. Weir. Driving air and circulating pump of condenser.
13578	,,	O Reynolds Arrangement of pump.
13709	,,	J. Taylor. Form of blades and of casing.
24430	,,	W. Mather. Arrangement of pump for well bores.
1235	1895	J. Gwynne. Automatic control of pump.
3024	,,	Hon. C. A. Parsons. Turbine-driven centrifugal pumps.
6155	,,	F. Pelzer. Form of blades.
12612	,,	P. Hövig. Arranging the pump casing in a water tank for excluding air, &c.
14457	,,	G. Mitchell. Form and arrangement of blades.
23751	,,	J. Gwynne and E. W. Sargeant. Form and arrangement of vanes, &c.
3666	1896	C. G. Pinette. General arrangement of pump.
7250	,,	J. E. Bousfield (La Société des Procédés Desgoffe et de Georges). Arrangement of vanes and discs.
14990	,,	H. Foster and A. Beresford. Arrangement of vanes and casing.
25468	,,	G. M. Capell. Form of blades or vanes.
3351	1897	J. Patterson. Forming a cutting surface on the blades.
4543	,,	K. Farkass. Control of pump by height of liquor in the delivery tank.
11238	,,	C. E. Beckman. Devices for taking up wear.
15968	,,	F. A. Robinson. Adjusting the suction and delivery passages.
17053	,,	J. Bernays. Priming arrangements.
21851	,,	A. F. Spooner (Soc. Emile Salmson et Cie). Controlling direct electrically-driven pump.
27089	,,	E. Crawshaw. Pump for use with chemicals.
28401	,,	H Förster. Arrangement of blades.
28944	,,	C. G. Pinette. Arrangement of pump.
4609	1898	S. C. Davidson, Shape of blades.
11392	,,	J. H. and F. Riley. Pump for corrosive liquids.
14926	,,	H. B. Barlow (Jennings Pump Proprietary). Pump for gritty liquids.
19030	,,	M. R. Ruble. Form of impeller.
19543	,,	S. C. Davidson. Arrangement of pump.
21380	,,	S C. Davidson. Supporting the vanes.
25988	,,	J. Gwynne and E. W. Sargeant. Priming arrangements.
26367	,,	S. C. Davidson. Arrangement of casing.
26776	,,	A. W. and L. W. Case. Arrangement of pump.

No.	Year.	Name of Patentee and Subject Matter.
27485	1898	F. Lobnitz. Protecting bearings of the pump.
838	1899	E. F. and W Marsh. Construction of pump for raising chemicals.
1616	,,	H. B. Barlow (J. Moore). Liner and casing of pump.
2209	,,	F. G. Lundwall. Pump in which blades rotate in an air vessel.
5452	,,	S. C. Davidson. Form of blades.
6829	,,	J. Gwynne and E. W. Sergeant. Pump for dredging and like purposes.
8030	,,	E. E. Marchand. Construction of delivery passage.
8547	,,	R. P. Park. Pump for dredging purposes.
9039	,,	H. A Thirion. Priming arrangements.
9658	,,	I. Smith. Construction of impeller and casing.
15973	,,	S. C. Davidson. Mounting pump casing.
19252	,,	S. C. Davidson. Mounting of blades.
22194	,,	W. L. Wise (Aktiebolaget de Lavals Angturbin). Centrifugal pump driven by a steam turbine.
24873	,,	S. C. Davidson. Forming blade carrier conical.
12298	1900	W. A. Granger. Forming delivery pipe tapered.
16068	,,	E. E. Marchand. Pump for propelling vessel by water jets.
18343	,,	E. Seitz. Arrangement of pump casing and of packing rings.
20216	,,	F. de Mare. Pumps for use with mercury.
8440	1901	A. Krank. Centrifugal pump driven by a steam turbine.
12116	,,	W. Wheatley (J. Richards). Forming blades on the exterior of the impeller.
17247	,,	W. and E. Allday (W. Hosken). Pump for muddy water.
18106	,,	A. C. E. Rateau. Pumps for high or low pressures.
18592	,,	C. Higgins (B. A. Smith and A. G. M. Michell). Driving a turbine wheel by the delivery from a centrifugal pump.
20268	,,	G. M. Capell. Cooling shaft and bearings of pump.
20466	,,	A. G. Bloxam (Gebrüder Sulzer). Varying height of lift of multiple centrifugal pumps.
21286	,,	S. C. Davidson. Construction of casing.
22406	,,	J. W. Hall. Driving of turbine pumps.
24944	,,	A. March. Centrifugal pump for use with acids.
10077	1902	J. D. McRae. Construction of pump.
10078	,,	J. D. McRae. Arrangement of centrifugal pump in a condensing system.
18099	,,	J. Hedlund. Packings for pump shafts.
18100	,,	J. Hedlund. Protecting the pump shaft.
21821	,,	S. C. Davidson. Mounting the pump casing.
24520	,,	A. Morcom and Belliss and Morcom. Preventing the escape of air past the packing of pump shaft.
26067	,,	Gebrüder Sulzer. Preventing access of air to vertical axis pumps.

No.	Year.	Name of Patentee and Subject Matter.
27387	1902	A. C. E. Rateau. Governing pump and regulating discharge.
28367	„	W. and E. Allday (W. Hosken). Taking up wear of impeller and of casing.
533	1903	C. H. Jaeger. Designing casing for fitting intermediate portions to make a multiple pump.
4673	„	C. H. Jaeger. Varying the area of the delivery channel.
6209	„	F. Janecek. Combined electric motor and centrifugal pump.
7479	„	P Kugel. Impeller and guides of high-pressure pump.
9007	„	E. W. Brooks. Form of impeller of centrifugal pump.
9501	„	R. Andrew. Centrifugal pump for dredging.
10257	„	Beaumonts Ltd. and W. and J. W. S. Beaumont. Arrangement of pumps and bearings.
12650	„	T. Reuter. Arrangement of pump in a condenser system.
18396	„	L. Vojacek. Form of impeller of pump.
20147	„	B. Jackson. Arrangement of series pumps and balancing end thrust.
20418	„	N. K. F. Hanson. Arrangement for varying delivery pressures.
25640	„	J. Richards Arranging a continuous channel around the impeller.
26510	„	R. T. Binnie. Form of impeller of pump.
28327	„	H. E. Newton (H. R. Worthington). Balancing and preventing leaking
28732	„	J. Murrie. Preventing foaming.
89	1904	Sulzer Gebrüder. Means for fixing parts accurately in position.
396	„	Sulzer Gebrüder. Guiding water from one set of vanes to the next.
3104	„	J. Tylor and Sons and A. P. Donnison Mounting pump for circulating water in internal combustion engines.
3308	„	L. Vojacek. Arranging suction orifice eccentrically to the driving shaft.
3579	„	H. E. Newton (H. R. Worthington). Varying discharge pressure.
5704	„	Warwick Machinery Co. (General Electric Co.). Arranging a centrifugal pump at the exhaust end of a steam turbine.
6196	„	H E. Newton (H. R. Worthington). Balancing pressures.
6707	„	P. M. Justice (E. Wilkinson). Combined centrifugal pump and steam turbine.
6708	„	P. M. Justice (E. Wilkinson). Combined centrifugal pump and steam turbine.
6709	„	P. M. Justice (E. Wilkinson). Combined centrifugal pump and steam turbine.

No.	Year.	Name of Patentee and Subject Matter.
8855	1904	E. Hopkinson and A. E. L. Chorlton. Providing a continuous flow through the pump and balancing end pressures.
14719	„	H. E. Newton (H. R. Worthington). Preventing leakage of water past the pump blades.
16397	„	O. Imray (A. F. Smulders). Centrifugal apparatus for separating water from sludge.
16592	„	Aktieselskabet Elling Compressor Co. Adjusting the guide vanes for efficiency.
21891	„	S. C. Davidson. Eccentric arrangement of suction orifice
21891A	„	S. C. Davidson. Arrangement of casing and of inlet passage.
22254	„	Sulzer Gebrüder. Arrangement and size of pump discs for convenience of building.
24686	„	S. M. Lillie. Centrifugal pumps for use in evaporating apparatus.
26418	„	L. Wilson. Centrifugal pumps for ship steering and propulsion.
27074	„	S. C. Davidson. Arrangement of inlet passage
28736	„	H. E. Newton (H. R. Worthington). Arrangement of pump for use in condensing system.
28978	„	J. E Evans-Jackson (Turbine Pump Co.). Shape of impelling and fixed blades of pump.
30	1905	J. M. Hewitt. Providing diffusing blades between the impellers.
3180	„	E. G. Harris. Regulating amount of discharge from pump, &c.
4674	„	H. E. Newton (H. R. Worthington). Balancing the pressures on the impellers.
7469	„	H. E. Newton (H. R Worthington). Construction of multi-stage centrifugal pump for running at high speeds.
7911	„	H. Ludewig. Arranging a turbine wheel to be driven by water delivered from the impeller of a centrifugal pump.
7999	„	A. C. E. Rateau. Governing arrangements of centrifugal pumps.
8452	„	Soc. Anonyme Westinghouse and M. Leblanc. Use of centrifugal pump in condensing system.
8454	„	Soc. Anonyme Westinghouse and M. Leblanc. Use of centrifugal pump in condensing system.
9840	„	Tangyes Ltd. and A. H. Yeandle. Arrangement of pump casing and blades.
10104	„	L. Beaudoin. Form of centrifugal pump for cooling liquids.
10684	„	S C. Davidson. Position of impeller relating to casing and to discharge passage.
10981	„	J. Y. Johnson (Jeanesville Iron Works Co.). Balancing end thrust on the rotating parts of the pump.

No.	Year.	Name of Patentee and Subject Matter.
12741	1905	F. W. Howarth (Aktiebolaget de Lavals Angturbin). Attachment to impeller and casing of pump for use with dirty liquids.
15175	,,	H E. Newton (H. R. Worthington). Mounting a number of impellers in a common suction passage with a common delivery pipe.
15581	,,	H Davey. Form of pump, and balancing arrangement.
15821	,,	A. E. Mohring and Crompton and Co. Priming arrangements.
16384	,,	E. S. Lea and J. Degen. Balancing the impellers against longitudinal thrust, &c.
16641	,,	H. E. Newton (H. R. Worthington). Form of the suction passage.
18676	,,	O. Kolb. Arrangement of centrifugal pump in a condensing system.
19470	,,	H. A. Kunzli. Balancing multi-stage pumps.
20664	,,	Hathorn, Davey and Co. Ltd., and F. G. Heseldin. Arrangements for preventing eddy currents in pumps.
21680	,,	N. K. F. Hanson. Form of centrifugal pump.
23923	,,	J. T. Rossiter. Devices for facilitating examination and repairs.
25391	,,	Société l'Éclairage Électrique. General arrangement of pump.
442	1906	C. H. Jaeger. Balancing thrust in centrifugal pumps.
2796	,,	Gebrüder Sulzer. Balancing thrust in centrifugal pumps.
3462	,,	J. Gwynne and E. W. Sargeant. Double inlet compound centrifugal pumps.
5547	,,	H. Tütola. General arrangement of a centrifugal pump.
18005	,,	C. Wedekind. Arranging pump wheels alternately on two parallel shafts.

INDEX.

R

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T

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V

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